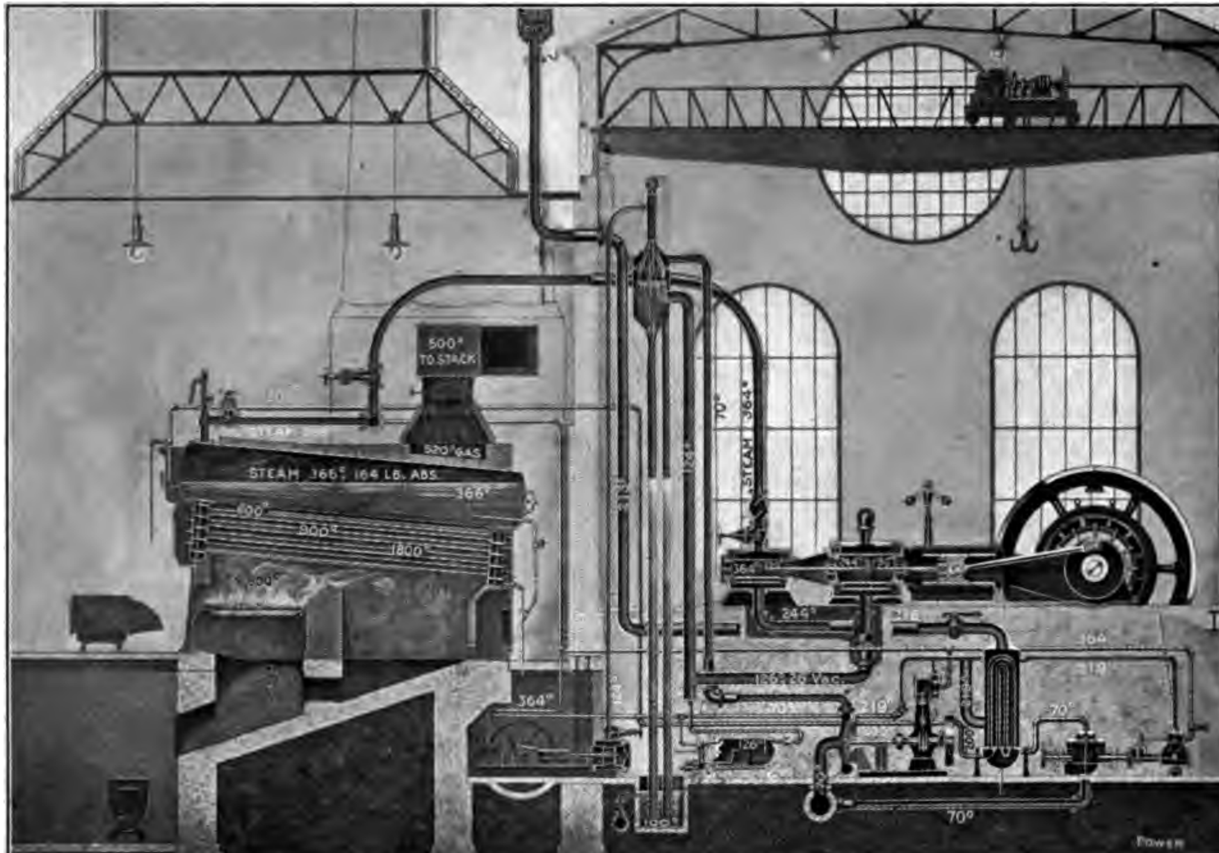
ENGINEERING OF POWER PLANTS



Gradations of Temperature in a Condensing Steam Plant

(Frontispiece)

ENGINEERING
OF
POWER PLANTS

BY

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PREFACE

This work is not a treatise on power plants.

It is simply an epitome of the subject arranged by the authors for convenient classroom use.

Originally compiled by one of the authors in 1908 for use at the Case School of Applied Science, these notes have been used for eight years at that institution and for four years at the University of Pennsylvania as the fundamental course in power plants for all senior engineering students, including mechanical, electrical, chemical, civil and mining engineers.

Besides offering much general material relating to the Engineering of Power Plants, the two underlying thoughts in the preparation of this text for classroom use have been—

(*a*) To bring to the student a realization of the fact that engineering, although based on the exact sciences, is not itself an exact science but requires, on the part of the successful engineer, a natural fund of "common sense" and the application of engineering judgment—a realization of the fact that *accuracy* may mean within *twenty per cent.* and not the seventh place to the right of the decimal point; and

(*b*) To give the student some idea of the commercial side of engineering—a field too seldom touched upon in many engineering courses.

The cost figures presented must be used with caution as market variations are such and local conditions so different that such data can be at best only approximate.

Although the authors have endeavored to give credit for data copied from various sources, much of the material has been so subdivided and used by so many different writers that it has not been feasible to trace it to its original source.

The authors desire to acknowledge their appreciation of the excellent service rendered by Paul J. Kiefer in checking the material presented in these notes.

R. H. F.
G. A. O.

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ENGINEERING OF POWER PLANTS

CHAPTER I

SOURCES OF ENERGY

For industrial purposes energy is derived from:

(a) Vital forces; muscular power of men and animals.
(b) Gravity; energy of the wind and flowing water.
(c) Chemical forces; energy of fuel.

In considering the various types of power, it is convenient to further subdivide item (c) into:

(d) Steam.
(e) Gas.
(f) Compressed air, hot water and refrigeration.
(g) Hydraulic power; with application to pumps, elevators, cranes, etc.
(h) Electric.

Unit of Power.—The unit[1] of power is the horsepower (hp.). One horsepower = 33,000 ft.-lb. per minute or 550 ft.-lb. per second in American and English practice. The metric horsepower used in Europe = 542.47 ft.-lb. per second or 0.9863 English horsepower. The metric horsepower is known as the pferde starke (P.S.) in German and cheval de vapeur in French. The unit used in electrical measurements of power is international and is called the watt, of which 746 correspond to the mechanical energy in 1 hp. The metric horsepower = 736 watts.

Efficiency of a Machine.—Kent states that the efficiency of a machine is a fraction expressing the ratio of the useful work to the whole work performed, which is equal to the energy expended. The limit to the efficiency of a machine is unity, denoting the efficiency of a perfect machine in which no work is lost. The difference between the energy expended and the useful work done, or the loss, is usually expended either in overcoming friction or in doing work on bodies surrounding the machine from which no useful work is received. Thus in an engine propelling a vessel part of the energy exerted in the cylinder does the useful work of giving motion to the vessel, and the remainder is spent in overcoming the friction of the machinery and in making currents and eddies in the surrounding water.

[1] In 1913, Mr. H. C. Stott suggested a double unit called the Myriawatt as a standard for steam-electric power. This standard has not yet been adopted. For discussion, see *Journal* A. S. M. E., February, 1913.

A common and useful definition of efficiency is "output divided by input."

Muscular Power of Men and Animals.—In dealing with living motors, it must be borne in mind that the motors are seldom alike, that the same motor varies at different times and that such motors can be used only intermittently as rest is absolutely essential.

The various treatises on the subject point out that such motors will differ with:

(*a*) The health of the specimen, his muscular development, his nervous temperament, his disposition, his degree of stimulus of interest and will.
(*b*) The species of animal, the race of man.
(*c*) The amount of practice, the degree of training.
(*d*) The abundance of food and air; the climate.

These items are all important but it is hardly possible to determine their exact effect upon the motor.

The more tangible elements that effect the work accomplished are:

(*e*) The relation of the working hours to those of rest in the 24 hr. of the day.
(*f*) The relation between the maximum exertion possible and the force actually exerted.
(*g*) The speed in feet per second.
(*h*) The nature of the machine receiving the effort of the motor force.

The effort of the application of energy should be to secure the greatest foot-pounds in a continuous day's work.

Labor of Men.—Man's labor is usually in lifting, pushing or pulling, or in transporting weights. Experimental data are as follows:

WORK OF MEN AGAINST KNOWN RESISTANCES (RANKINE)

Kind of exertion	Resistance, lb.	Velocity, ft. per sec.	Hours per day	Ft.-lb. per sec.	Ft.-lb. per day (8 hr.)
1. Raising his own weight up stair or ladder	143.0	0.5	8	71.5	2,059,200
2. Hauling up weights with rope and lowering the rope unloaded	40.0	0.75	6	30.0	648,000
3. Lifting weights by hand	44.0	0.55	6	24.2	522,720
4. Carrying weights upstairs and returning unloaded	143.0	0.13	6	18.5	399,600
5. Shoveling up earth to a height of 5 ft. 3 in.	6.0	1.3	10	7.8	280,800
6. Wheeling earth in barrow up slope of 1 in 12½, horiz. veloc. 0.9 ft. per sec., and returning unloaded	132.0	0.075	10	9.9	356,400
7. Pushing or pulling horizontally (capstan or oar)	26.5	2.0	8	53.0	1,526,400
8. Turning a crank or winch	12.5	5.0	?	62.5	
	18.0	2.5	8	45.0	1,296,000
	20.0	14.4	2 min.	288.0	
9. Working pump	13.2	2.5	10	33.0	1,188,000
10. Hammering	15.0	?	8?	?	480,000

NOTE.—See TAYLOR's study on "Handling Pig at Bethlehem," "Principles of Scientific Management," Harpers, 1911, p. 25.

Records show that men have pulled once 182, 208, 227 and 267 lb. Ordinarily, at a crank a man will exert from 15 to 18 lb. continuously and from 25 to 30 or even 40 if applied intermittently. Such a crank will turn from 26 to 30 revolutions per minute.

At 18 lb., the foot-pounds per day are 1,296,000, while for an engine horsepower they are 15,840,000, or the man has one-twelfth or one-thirteenth the power of an engine horsepower.

Clark says that the average net daily work of an ordinary laborer at a pump, a winch, or a crane may be taken at 3300 ft.-lb. per minute, or 1⁄10 hp., for 8 hr. a day; but for shorter periods from four to five times this rate may be exerted.

Rowing in races is calculated to exact 4,000 lb. raised 1 ft. high per minute, or nearly 1⁄8 hp.

The action of the heart limits the stress which can be put on the human organism. For a short time like 2½ min., records of 11,550 ft.-lb. per minute have been recorded, but usually 3,000 ft.-lb. per minute is too high for acceptable service.

PERFORMANCE OF MEN IN TRANSPORTING LOADS HORIZONTALLY (RANKINE)

Kind of exertion	Load, lb.	Velocity, ft.-sec.	Hours per day	Lb. conveyed. 1 ft. per sec.	Lb. conveyed, 1 ft. per day
11. Walking, unloaded, transporting his own weight.	140	5.0	10	700.0	25,200,000
12. Wheeling load L in 2-whld. barrow, return unloaded....................................	224	1⅔	10	373.0	13,428,000
13. Wheeling load L in 1-wh. barrow, return unloaded....................................	132	1⅔	10	220.0	7,920,000
14. Traveling with burden........................	90	2½	7	225.0	5,670,000
15. Carrying burden, returning unloaded..........	140	1⅔	6	233.0	5,032,800
16. Carrying burden, for 30 sec. only............	252	0.0	...	0.0	
	126	11.7	...	1,474.2	
	0	23.1	...	0.0	

Coignet's apparatus was a hoist, in which dirt on one platform was lifted out of an excavation by the descent of laborers on a similar platform attached to rope passing over a pulley at the top. The laborers lifted their weight out of the excavation by climbing up ladders, and their descending weight over-balanced the material to be lifted. Brakes controlled the speed.

Where weights are lifted by men hauling over pulleys, about 40 lb. is an average pull.

Emerson states that to spade up a section of land would take an active man's energy for 500 years. With oil power tractors and gang plows three men can turn over 640 acres of land in 36 hr. It is good hard work to make a broad jump of 20 ft. at a speed of 10 miles an hour, rising 4 ft., from the ground, but aeroplanes at the international contest this year (1912) will fly 80 miles at a speed which may reach 110 miles an hour, and fly as easily at an altitude of 5,000 ft. as at 50. Formerly a man could carry a maximum load of 100 lb.; today his trains drag 6,000 tons and his ships carry 30,000 tons.

As pointed out by Greenfield[1]

"One of the serious objections to the use of man-energy for motive purposes lies in the impracticability of securing large amounts of power from even large groups of men. A little calculation will make this point clear: The power plant of a modern "department store" may contain, let us say, eight steam boilers with an aggregate capacity of some 4,800 hp. To produce 5,000 hp. by the use of men it would be necessary to employ $5{,}000 \times 10 = 50{,}000$ men, who are supposed to drive the electric generators, treadwheel fashion. Assigning a floor space of 2 ft. by 4 ft. to each man, these workers would require a total floor space of $50{,}000 \times 8 = 400{,}000$ sq. ft., which is about one-fifth of the total floor space in the Philadelphia store of John Wanamaker."

This point will be more fully emphasized by the following problem.

1. The total indicated horsepower of the turbine steamer Mauretania is 70,000. If the average demand is 60 per cent. of her rated horsepower, how many men would be required for a trans-Atlantic trip if man-power were substituted for the turbines on this steamer?

The most efficient way to use a man's effort is to have him lift his own weight, either by treadmill, tread-power or by using his weight as a counterpoise for the dead weight to be lifted. Here he does 4,000 ft.-lb. per minute as a counterweight or in treadmills, 3,000 ft.-lb. 2,600 to 2,750 ft.-lb. is a figure for use with a crank.

A heavy flywheel (200–400 lb.) is very useful in machines for human motors to equalize the unequal effort in parts of the revolution where the man can exert but little power.

All things considered a man seems to be most efficient for continued service when he works one-third of his day of 24 hr. at a speed of one-third of his maximum and exerts one-third of his maximum force.

Animal Motors.—

WORK OF HORSES AGAINST A KNOWN RESISTANCE (RANKINE)

Kind of exertion	Resistance, lb.	Velocity, ft. per sec.	Hours per day	Ft.-lb. per sec.	Ft.-lb. per day
...ering and trotting, drawing a light rail-...age (thoroughbred)	min. 22½ mean 30½ max. 50	14⅔	4	447½	6,444,000
... cart or boat, walking (draught-...)	120	3.6	8	432	12,441,600
... a gin or mill, walking	100	3.0	8	300	8,640,000
... a gin or mill, trotting	66	6.5	4½	429	6,950,000

[1] ...agazine, 1911, "Human Energy as a Motive Power" by B. S. GREEN-

Kent states that the average power of a draught-horse, as given in line 2 of the above table, being 432 ft.-lb. per second, is 432/550 = 0.785 of the conventional value assigned by Watt to the ordinary unit of the rate of work of prime movers. It is the mean of several results of experiments, and may be considered the average of ordinary performance under favorable circumstances.

PERFORMANCE OF HORSES IN TRANSPORTING LOADS HORIZONTALLY (RANKINE)

Kind of exertion	Load, lb.	Velocity, ft. per sec.	Hours per day	Transport per sec.	Transport per day
5. Walking with cart, always loaded	1,500	3.6	10	5,400	194,400,000
6. Trotting with cart, always loaded	750	7.2	4½	5,400	87,480,000
7. Walking with cart, going loaded, returning empty; *V*, mean velocity	1,500	2.0	10	3,000	108,000,000
8. Carrying burden, walking	270	3.6	10	972	34,992,000
9. Carrying burden, trotting	180	7.2	7	1,296	32,659,200

This table has reference to conveyance on common roads only, and those evidently in bad order as respects the resistance to traction upon them.

A horse towing or drawing at a walk will average a pull of 120 lb. at 3½ ft. per second or 2.3 miles per hour. At a trot the pull will be but 66 lb. at 6½ ft. per second.

In a whin or gin at a brisk walk, or at 3 ft. per second, the pull will be about 100 lb. The curved track lowers the efficiency of the draft. Forty feet is the usual diameter of circle for large machines. In towing the average figures are:

Miles per hour	Hours per day	Load in tons	Miles per hour	Hours per day	Load in tons
2.5	11.5	520.0	6.0	2.0	30.0
3.0	8.0	243.0	7.0	1.5	19.0
3.5	5.9	153.0	8.0	1.12	13.0
4.0	4.5	102.0	9.0	0.75	9.0
5.0	2.9	52.0	10.0	0.55	6.3

In horse power or horse gears the slow speed of the motor requires multiplying mechanism for high-speed machinery. The losses here limit the field for these motors.

Although the engine horsepower is 33,000 ft.-lb. per minute, the living horse does not keep this rate up all day. In pumping with horses the records are:

23,412 ft.-lb. per 8 hr. per day.
24,360 ft.-lb. per 6 hr. per day.
27,056 ft.-lb. per 4.5 hr. per day.
32,943 ft.-lb. per 3 hr. per day.

The average is from 21,000 to 25,000 ft.-lb. per minute.

Other animals used for draught purposes are the ox, the mule, the ass, the elephant, the reindeer and the dog. The ox-power is about 12,000 ft.-lb. per minute, the mule 10,000; the ass 3,500. Rankine favors two-thirds speed and same load for the ox as compared with the horse; for the mule one-half load and same velocity; for the ass one-quarter load and same velocity as for the horse.

For transporting burdens the camel, the dromedary and the llama may be added to the list.

The load of a freight camel is 550 lb. carried 30 miles per day for 4 days; for dromedary the load is 770 lb.; the llama, 110 lb.

Animal motors are used in frontier or colonial conditions for farming and forest service; for crushing and grinding sugar, for cotton work, for sawing, pumping, irrigation, etc.[1]

GRAVITY—ENERGY OF WIND AND WATER

Windmills.[2]—Horizontal-shaft four-sailed windmills of the "Dutch" type have been used for many years. A few vertical shaft wheels have also been used. These types, although efficient, were costly and hard to control and have been superseded by the "American" type which has spread over the entire globe.

If

P = wind pressure in pounds per square foot,

V = wind velocity, miles per hour, then from Stanton's experiments,

$$P = 0.0036V^2$$

If

W = work in foot-pounds per second, then for European mills of the Dutch type,

$W = 0.0011V^3$ (Coulomb)

This value of W is likely to be higher than shown by average practice.

For mills of the American type

$W = 0.00045V^3$ (Wolff)

$W = 0.00050V^3$ (Griffiths)

[1] See A.S.M.E., vol. 14, p. 1014, paper by Thomas H. Briggs, "Haulage by Horses."

[2] Note.—Among early experimenters the work of Smeaton and Coulomb stands out conspicuously. Excellent theoretical treatments will be found in "Windmills" by A. R. Wolff and in the Encyclopedia Britannica, 11th edition by W. C. Unwin.

See also "Modern Tests" in *Water Supply Papers* Nos. 1 and 8, U. S. Geological Survey. For wind-pressure formulas, see Stanton in *Proceedings* I.C.E., vol. 156 and works of Eiffel and Hagen.

If

D = diameter of wheel in feet,

hp. = horsepower,

then

$$\text{hp.} = \frac{D^2V^3}{853{,}000}$$

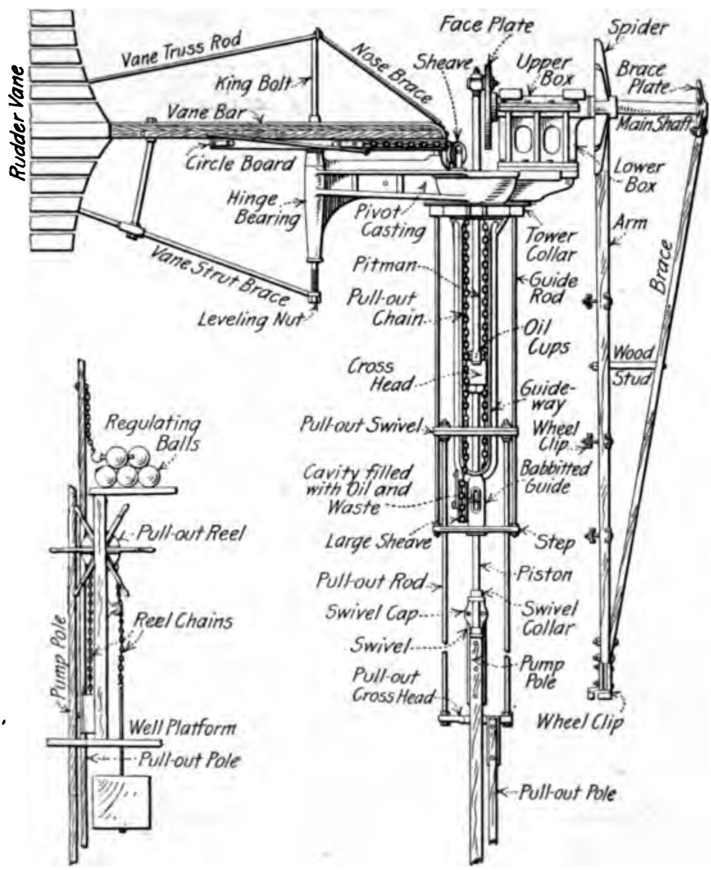


FIG. 1.

The efficiency of modern wheels varies from 5 to 30 per cent., 10 to 18 per cent. representing standard practice. Much of the power is lost in the gearing.

The best wind velocity is about 15 miles per hour, the useful range being from 10 to 20 miles.

The best wheel diameter seems to be about 12 ft., although wheels from 8 to 16 ft. in diameter give excellent results; 80-ft. wheels have, however, been successfully used.

A few wide blades are more efficient than many narrower ones. The vanes should not cover more than from 75 to 80 per cent. of the wheel area for best results and should never overlap.

The governing of windmills is done almost entirely by putting the wheel into the plane of the wind. Certain foreign wheels govern by feathering the vanes and some American wheels by swinging sections of vanes into the plane of the wind.

Windmills should be erected on towers or other elevated structures, but should not be set on hilltops or windy headlands.

Reports of the U. S. Weather Bureau show that the useful hours (those with V between 10 and 20) are rarely over 3,000 per year in wooded locations, but in the Western plains and in certain favored localities in the East they may be much higher.

Pumping is the best method of utilizing wind-power but many windmills are geared to run feed cutters and other agricultural machinery.

Successful American manufacturers are reported by Wolff to be meeting the following guarantees.

Capacity of Windmills.—

Designation of mill	Vel. of wind, in miles per hour	Revolutions of wheel per minute	Gallons of water raised per minute to an elevation of						Equivalent useful hp. developed	Average No. of hours per day during which this results will be obtained
			25 ft.	50 ft.	75 ft.	100 ft.	150 ft.	200 ft.		
Wheel, ft.										
8¼	16	70 to 75	6.162	3.016	......	......	......	......	0.04	8
10	16	60 to 65	19.179	9.563	6.638	4.750	......	......	0.12	8
12	16	55 to 60	33.941	17.952	11.851	8.485	5.680	......	0.21	8
14	16	50 to 55	45.139	22.569	15.304	11.246	7.807	4.998	0.28	8
16	16	45 to 50	64.600	31.654	19.542	16.150	9.771	8.075	0.41	8
18	16	40 to 45	97.682	52.165	32.513	24.421	17.485	12.211	0.61	8
20	16	35 to 40	124.950	63.750	40.800	31.248	19.284	15.938	0.78	8
25	16	30 to 35	212.331	106.964	71.604	49.725	37.349	26.741	1.34	8

In comparing Wolff's figures with those given by the formula it should be remembered that the values for horsepower as given in the table are overall horsepowers and include pump and pipe friction while the formula gives the horsepower available at the wheel shaft.

For windmill economy, Wolff gives:

Economy of Windmills.—

Designation of mill	Gal. of water raised 25 ft. per hour	Equivalent actual useful hp. developed	Avg. No. of hours per day during which this quantity will be raised	Expenses for actual useful power developed, in cents, per hour: For interest on 1st cost (1st cost including cost of windmill, pump and tower 5 per cent. per annum)	For repairs and depreciation (5 per cent. of 1st cost per annum)	For attendance	For oil	Total	Expense per hp., in cents, per hour
Wheel, ft.									
8½	370	0.04	8	0.25	0.25	0.06	0.04	0.60	15.0
10	1,151	0.12	8	0.30	0.30	0.06	0.04	0.70	5.8
12	2,036	0.21	8	0.36	0.36	0.06	0.04	0.82	5.9
14	2,708	0.28	8	0.75	0.75	0.06	0.07	1.63	5.8
16	3,876	0.41	8	1.15	1.15	0.06	0.07	2.43	5.9
18	5,861	0.61	8	1.35	1.35	0.06	0.07	2.83	4.6
20	7,497	0.79	8	1.70	1.70	0.06	0.10	3.56	4.5
25	12,743	1.34	8	2.05	2.05	0.06	0.10	4.26	3.2

Based on the figures quoted by Wolff in the fifth column the cost of the installations including windmill, pump and tower is approximately:

Wheel, ft.	Cost, $
8.5	145
10.0	175
12.0	210
14.0	435
16.0	670
18.0	790
20.0	1,000
25.0	1,200

AMERICAN WINDMILL AND TOWER PRICES

	Diam., ft.	Mill alone: Rated hp.	Rated r.p.m.	Weight machinery, lb.	Price f.o.b., Chicago	Tower: Height, ft.	Weight, lb.	Price f.o.b., Chicago
Wooden mill..	8½	0.08	35	400	$28.50	20	203	$15.60
	10	0.12	35	500	33.00	30	500	30.00
	12	0.25	30	700	42.00	40	950	50.00
	14	0.40	28	1,010	75.00	40	1,300	82.50
	16	0.55	25	1,685	114.00	40	1,450	92.00
	18	0.75	22	1,880	128.00	40	1,450	92.00
	20	1.00	20	2,990	194.00	40	2,935	186.00
	25	1.25	16	4,300	352.00	40	2,935	186.00
Steel mill.....	8	0.08	30	325	22.50	20	203	15.60
	10	0.12	25	500	36.25	30	500	30.00
	12	0.25	20	950	66.00	40	950	50.00
Steel mill.....	8	.....	....	....	30.00	25	435	26.50
	10	.....	....	....	50.00	25	435	26.50
	12	1.5–3	....	....	70.00	25	585	38.50
	14	2–4	....	....	120.00	60	1,435	88.00
	16	3–6	....	....	175.00	60	2,550	160.00
	20	.....	....	....	360.00	60	5,400	410.00

Bulletin No. 105 of the North Dakota Agricultural Experiment Station contains a description of a 1.4-kw. electric light and power plant deriving its power from a 16-ft. aeromotor windmill on a 20-ft. wooden tower. In connection with the generator a 62-cell 40-amp.-hr., 110-volt Planté type storage battery is used.

Cost of 16-ft. wheel, tower and governing pulley	$200.00
Cost of house	35.00
Cost of dynamo 1.4 kw., 150 volts, 1,800 r.p.m	110.00
Cost of battery	550.00
Cost of switchboard	150.00
	$1,045.00
Yearly charges:	
Interest, 6 per cent	$62.70
Dep. 10 per cent	104.50
Attendance	12.80
Oil	5.00
Repairs	....
	$185.00

Year 1912—kilowatt-hours at switchboard per year 3,300. Cost per kilowatt-hour = 5.6 cts.

This wheel gave a maximum of 1,009 hp.-hr. in April and a minimum of 332 hp.-hr. in August.

The two important factors in the success of this installation were the governor and an automatic regulator which cut the battery out and into the circuits as required. The attendance charged to the apparatus seems very small and the absence of repair charges is remarkable.

Tide and Wave Motors.—Albert W. Stahl, U.S.N., finds[1] the energy of ocean waves to be as follows:

TOTAL ENERGY OF DEEP-SEA WAVES IN TERMS OF HORSEPOWER PER FOOT OF BREADTH

Ratio of length of waves to height of waves	Length of waves in feet							
	25	50	75	100	150	200	300	400
50	0.04	0.23	0.64	1.31	3.62	7.43	20.46	42.01
45	0.05	0.29	0.79	1.62	4.47	9.18	25.30	51.94
40	0.06	0.36	1.00	2.05	5.65	11.59	31.95	65.58
35	0.08	0.47	1.30	2.68	7.37	15.14	41.72	85.63
30	0.12	0.64	1.77	3.64	10.02	20.57	56.70	116.38
25	0.16	0.90	2.49	5.23	14.40	29.56	80.85	167.22
20	0.25	1.44	3.96	8.13	21.79	45.98	126.70	260.08
15	0.42	2.83	6.97	14.31	39.43	80.94	223.06	457.89
10	0.98	5.53	15.24	31.29	86.22	177.00	487.75	1,001.25
5	3.30	18.68	51.48	105.68	291.20	597.78	1,647.31	3,381.60

[1] See "The Utilization of the Power of Ocean Waves," *Transactions* A.S.M.E., vol. 13, p. 438.

Commenting on the practical utilization of this form of energy he divides the subject into:

1. The various motions of the water which may be utilized for power purposes.
2. The wave-motor proper—that is, the portion of the apparatus in direct contact with the water, and receiving and transmitting the energy thereof; together with the mechanism for transmitting this energy to the pumping or other suitable machinery for utilizing the same.
3. Regulating devices, for obtaining a uniform motion from the more or less irregular and variable action of the waves, as well as for adjusting the apparatus to the state of the tide and condition of the sea.
4. Storage arrangements for ensuring a continuous and uniform output of power during a calm or when the waves are comparatively small.

Taking up first the consideration of the motions that may be utilized for power purposes, we find the following:

1. Vertical rise and fall of particles at and near the surface.
2. Horizontal to-and-fro motion of particles at and near the surface.
3. Varying slope of surface of wave.
4. Impetus of waves rolling up the beach in the form of breakers.
5. Motion of distorted verticals.

Mr. Stahl further states:

"Possibly none of the methods described in this paper may ever prove commercially successful; indeed the problem may not be susceptible of a financially successful solution. My own investigations, however, so far as I have yet been able to carry them, incline me to the belief that wave-power can and will be utilized on a paying basis."

Wave motors are of two forms, the paddle type and the float type. In the former the paddle swings backward and forward with the wave motion, moving a shaft or pair of shafts by rachets. In the latter a heavy float is lifted by the waves and in falling drives a shaft by means of rachets.

Many attempts in this direction have been made but no successful machine has been developed that can withstand the tremendous power of severe storms.

A float-type motor erected near Los Angeles, Cal., developing sufficient power for about 30 incandescent electric lights, withstood the storms for one year. In this locality the waves, though high are regular. A storage battery was used for storing the energy developed during the day.

There are several tide mill ponds along the Connecticut and New York shores. One such pond at Stamford is reported to have an area of some 10,000,000 sq. ft. One writer on the subject estimates that about 50 hp.-hr. can be realized per million square feet of pond area.

A plant that attracted much attention at the time of its construction is the hydraulic air compressor of the Rockland Power Co., Rockland, Maine. The plant is described as follows:

"The plant consists of two basins, a high- and a low-water one, as in the Decoeur system. Each basin has an area of 1 sq. mile and there is a tide of 10 ft. From the high-water basin a 15-ft. shaft extends vertically downward for 203 ft., and then is connected by a horizontal tunnel to a 35-ft. shaft extending upward to the low-water tank. At the top of the down-flow shaft there are 1,500 half-inch air inlet tubes, through which air is drawn into the water and carried to the bottom of the shaft. The air separates at the bottom and accumulates in an air chamber while the water flows up the larger shaft to the low-water tank. The air is under a head of water of 195 ft. and is piped to the surface through a 14-in. pipe which joins a 30-in. main. This apparatus develops 5,000 hp. and has an efficiency of 75 per cent. It has no moving parts to break or get out of order. Air compressed by this method is very dry, being about three times as dry as the atmosphere and this is a decided advantage as the pipe resistance of dry air is very low and velocities as high as 70 ft. per second may be used. Dry air also can be used cold for expansion purposes and will not freeze. As an actual test an 80-hp. Corliss engine was run for 10 hr. on this air, the admission temperature being 5.3°F. and the exhaust −40°F. After the run there was no trace of frost in the exhaust port or passage. The only cost in connection with the operation of this compressor is the salary of a watchman to keep ice, timber, etc., from entering the inlet shaft. The construction cost of this particular plant amounted to about $100 per horsepower. Each basin of 1 acre area and with a tide of 9 ft. can produce about 5 hp. A basin with an area of 200 acres and so located that it would not require more than 3 ft. of dam per acre would be a commercial success if developed into an air-compressing plant."

Apparently the most successful tide machines have taken advantage of the rising tide for storing water in tanks or basins from which it is passed through waterwheels or turbines.

Solar Engines.[1]—Using Herschell's data, Ericsson about 1870 estimated the direct heat energy of the sun in 45° latitude to be equivalent to 13,000,000 hp. per square mile. Buchanan in Egypt in 1882 by means of better apparatus recorded the extremely high rate of 3,245 B.t.u. per square foot per minute, which is equivalent to 214,000,000 hp. per square mile.

Solar engines are not direct in their action, but use steam as an intermediary. The main interest is, therefore, in the boiler.

Monehart's apparatus, exhibited in Paris in 1878, consisted of a conical mirror 112 in. in diameter, which concentrated the sun's rays on a boiler containing about 44 lb. of water. With 45 sq. ft. of reflecting surface he succeeded in evaporating 11 lb. of water from a feed-temperature of 68°F. into steam at 75 lb. pressure.

[1] See *Engineering News*, May 13, 1909 and Nov. 17, 1910.

Ericsson's apparatus consisted of a parabolic mirror 11 by 16 ft. which concentrated the sun's rays on a boiler 6¼ in. in diameter and 11 ft. long. This machine, erected in New York in 1883, developed about 3.25 hp. in a 6-in. diameter by 8-in. stroke engine.

The steam was evaporated at 20 lb. gage pressure. A good condenser was used with this plant.

Many similar machines have been built with varying success. Most of these machines were mounted on equatorial mountings and were swung to meet the sun but fixed boilers or rather evaporators have been proposed. Shuman in 1907 built at Tacony, Pa., an experimental plant, each unit of which consisted of a sheet-iron boiler 3 ft. square with a

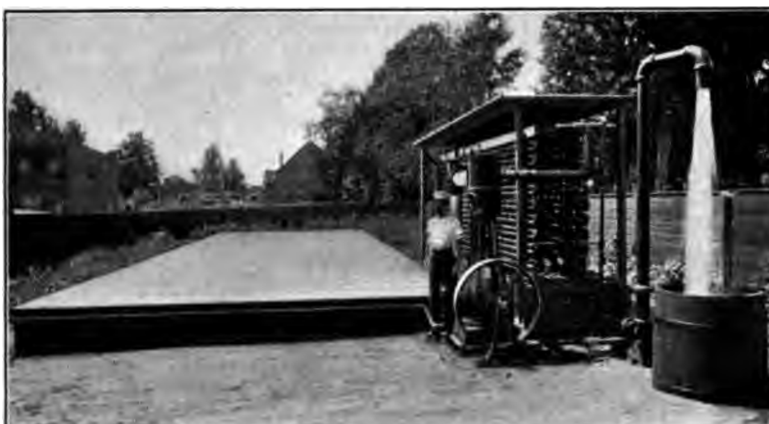


FIG. 2.—Original Shuman sun power plant, Aug., 1907, Tacony, Pa.

water space ⅛ in. wide. This is placed on a table with a mirror on each side. The plant contained 26 banks of 22 units each, a total of 5,000 sq. ft. of boilers and 5,300 sq. ft. of mirrors, 10,300 sq. ft. in all. 4,800 lb. of water were evaporated in 8 hr. of sunshine at atmospheric pressure. A special engine was designed to work under this low range of pressures and from 20 to 32 hp. was developed. This plant was tested by Prof. R. C. Carpenter (see *Engineer*, London, July 5, 1912).

Later Shuman in his plant at Meadi, Cairo, Egypt, went back to the moving parabolic mirrors with a 15-in. wide flat cast-iron boiler in the focus. This plant has five reflectors 204 ft. long, 13¼ in. wide with a thermostat control to keep the axis in the sun's plane. The total reflecting surface is 19,000 sq. ft. This plant averages 1,100 lb. of steam evaporated per hour for 10-hr. day.

The engine at Tacony was a 24 by 24-in. special engine with extra large exhaust valves running 12–150 r.p.m. The Cairo engine was a

36 by 36-in. running at 110 r.p.m. The Cairo plant cost, exclusive of land, $7,600 or $140 per brake horsepower.

Shuman figures that sun-power is economical in Egypt when coal costs more than $2.40 per ton.

In the Shuman invention a tract of land is rolled level, forming a shallow trough. This is lined with asphaltum pitch and covered with about 3 in. of water. Over the water about 1/6 in. of paraffine is flowed, leaving between this and a glass cover about 6 in. of dead air space. It is estimated that a power plant of this type to cover a heat-absorption area of 160,000 sq. ft., or nearly 4 acres, would develop about 1,000 hp. Provision is made for storing hot water in excess of the requirements of a low-pressure turbine during the day, to be utilized for running the turbine during the period when there is no absorption of heat. The heated water is run from the heat absorber to the storage tank, thence to the turbine, through a condenser and back to the heat absorber. The water enters the thermally insulated storage tank, or the turbine, at about 202°F. With a vacuum of 28 in. in the condenser, the boiling point of the water is reduced to 102°, and as it enters the turbine nearly 10 per cent. explodes into steam. Mr. Shuman estimates that a 1,000-hp. plant built upon his plan would cost about $40,000.

Willsie's plant at the Needles, Cal., built in 1908–09, utilizes the sun's heat to vaporize sulphur dioxide through the medium of water heated in a pipe coil encased in glass. His apparatus developed 20 net hp.

He figures his apparatus to cost as follows:

400-hp. plant, per horsepower	
Solar heater	$100.00
Heat storage plant (100 hr.)	10.00
Engine and pumps	20.00
SO_2 vaporizers	15.00
Condenser	15.00
Emergency boiler	2.75
SO_2	1.25
	$164.00
Operating cost per horsepower hour	
Interest dep., etc	0.19
Labor	0.27
Supplies	0.15
	0.61 cts.

Willsie compares the cost per horsepower hour in a 400-hp. steam-electric and solar-electric power plant, and finds that the steam plant would have to obtain its coal for $[illegible] a ton to compete with the sun-power plant in districts favorable to the latter.

The following table presents a brief summary of the development of the sun motor to date.

Year	Inventor	Location	Reflecting surface, sq. ft.	Water evaporated per hour, lb.	Square foot of reflecting surface per pound of evaporation
1878	Mouchot	Paris	45	11	4.09
1883	Ericsson	New York	162	*	
1911?	Shuman	Tacony	10,300	600	17.16
1913	Shuman	Cairo	19,000	1,100	17.28

* 3.25 hp. developed in engine.

The efficiency of the evaporative apparatus depends on the quality of the heat insulation. Ericsson's plant was much better in this regard than any of the others. Shuman's Tacony plant was nearly twice as good as the Cairo plant but was more costly.

All modern plants use the steam at or near the atmospheric pressure and must have plenty of condensing water for the vacuum. It would appear that the turbine is the best form of apparatus to use steam between these limits.

Prof. Fessenden's proposition (see British Association Adv. Science, 1912) including windmills, turbines, a 1,000-ft. well, exhaust turbines and flat solar heaters, has not as yet been experimentally tried.

Energy of Fuel.—In coal and other fuels an enormous capacity for doing work is stored in very compact bulk. It is liberated from the fuel gradually as required, and the limits of the available quantity have not yet been reached, although the time limit on anthracite coal and possibly other fuels appears to be close at hand.

Such fuels are to be had in nearly all regions and where they are not native they are easily transported. If desired the energy resident in them can be transported in the form of gas to the place where it is to be used.

It should not be forgotten that but a very small percentage of the heat energy in the fuel is actually converted into useful work at the machine. Roughly, if the fuel be used in a steam plant, 30 per cent. of the heat is lost by radiation and up the stack. Of the possible 70 per cent. that goes ✓ to the engine, nearly 90 per cent. is carried away by the condensing water or dissipated in the exhaust. Of the 10 per cent. or less converted into work a portion is used in overcoming friction, so that the useful work at the machine or busbars represents, in efficient steam plants, between 5 per cent. and 10 per cent. of the heat energy of the fuel thrown into the furnace under the boiler.

In the latest large unit installations efficiencies of 17 and 18 per cent. have been reached, but these are very exceptional.

Similar losses, although not necessarily of the same magnitude are evident in all present methods of power development from fuel.

Analysis of Development in a Power Plant.—Hutton states that in the typical power plant there are five steps:

1. Generation or liberation of the stored or accumulated energy.
2. The storage or accumulation of the energy of heat thus liberated from the fuel in a suitable vessel or reservoir from which it can be drawn off as required. (In the steam plant this is the boiler.)
3. The appliance whereby the energy stored in the boiler as potential energy is transformed into actual energy by being made to exert force through a prescribed path under the control of capable intelligence. (This is the engine.)
4. The controlled force acting through the controlled space or path is to be transmitted from the engine or prime mover to the machine or apparatus which is to be driven. (This gives rise to mechanism and transmission machinery.)
5. The industrial work of manufacturing, propelling or whatever may be the function of the generated power, is the last link in the chain.

In water-power plants the liberation or storage of energy is done for the engineer before his work begins. This is also true of the windmill motor. In the gas, hot-air, or direct-combustion engine there is no storage step in the process, but the energy must be utilized as fast as it is released. On the other hand, for the gas-engine plant which produces its own gas there is a step of accumulation of energy which is lacking when solid fuel is burned directly under the boiler.

D. B. Rushmore has estimated the amount of power used in the United States as follows:

	Horsepower
Horses and mules	25,000,000
Automobiles	25,000,000
Steam and naval vessels	5,000,000
Steam railroads	50,000,000
Irrigation	500,000
Mines and quarries	6,000,000
Flour, grist and saw mills	1,250,000
Manufactures	25,000,000
Central stations	8,000,000
Isolated plants	4,250,000
Electric railways	4,000,000
Total	154,000,000

Excluding the first four divisions, it is evident that about 49,000,000 hp. are used in the ordinary manufacturing, heating, lighting and electric railway business. Of this power at least 25,000,000 hp. or over 50 per cent. is used in the form of electric energy. It is probable that nearly

30,000,000 hp. are produced by steam; 12,000,000 hp. by water and about 7,000,000 hp. by gas or oil motors.

The transmission of power in its various fields of electrical transmission, compressed air, high-pressure water, shafting, belting, gearing or linkage is an extensive subject in itself, but will be touched upon briefly later in these notes..

Although the reciprocating steam engine may be regarded by many as obsolescent, it holds an important place in power production. Its mechanism and method of operation are better known by the average engineer than those of the more recent forms of prime movers. It is, therefore, used in these notes as a basis for the development of the essential principles of a power plant.

CHAPTER II

THE STEAM ENGINE

Horsepower of a Cylinder.—

Let P = pressure on piston in lb. per square inch.
= mean effective pressure (m.e.p.).
L = length of stroke in feet.
A = area of piston in square inches.
N = number times per minute that piston is acted upon by the pressure.

then $\text{hp.} = \dfrac{PLAN}{33{,}000}$.

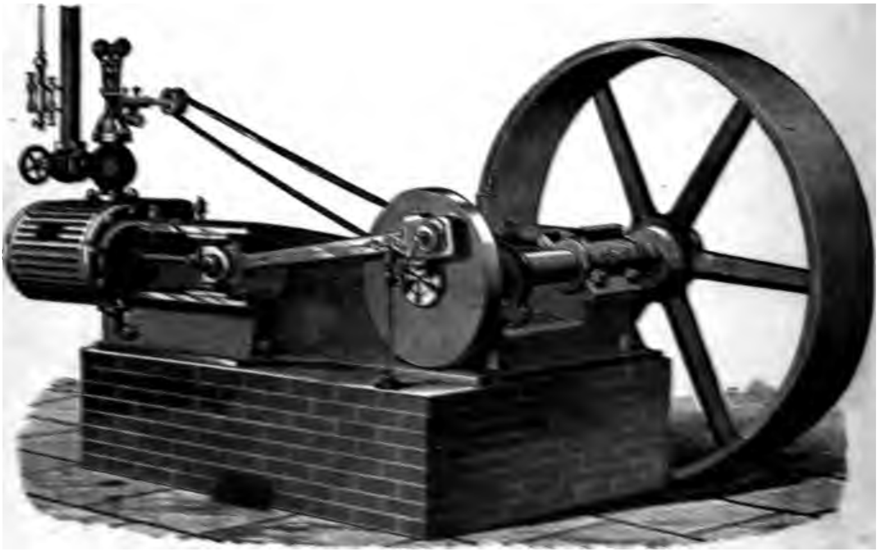


FIG. 3.—Simple slide-valve engine, throttling governor.

In an engine once constructed A and L are fixed or constant and the factor 33,000 is constant. If then K denotes the fraction $\dfrac{AL}{33{,}000}$ the hp. formula may be written hp. = PNK in which K is called the engine constant.

It is not necessary that the piston travel back and forth in a straight ne, as in reciprocating engines, although this is the common type. When the piston or area receiving the steam pressure travels in a circular path continuously in the same direction the engine is called a rotary steam engine.

The following handy rule is given for estimating the horsepower of a single-cylinder engine.

Square the diameter and divide by 2. This is correct whenever the product of the mean effective pressure and the piston speed (in feet per minute) = 21,000.

viz., when m.e.p. = 30 and S = 700.
m.e.p. = 33 and S = 600.
m.e.p. = 38 and S = 550.
m.e.p. = 42 and S = 500.

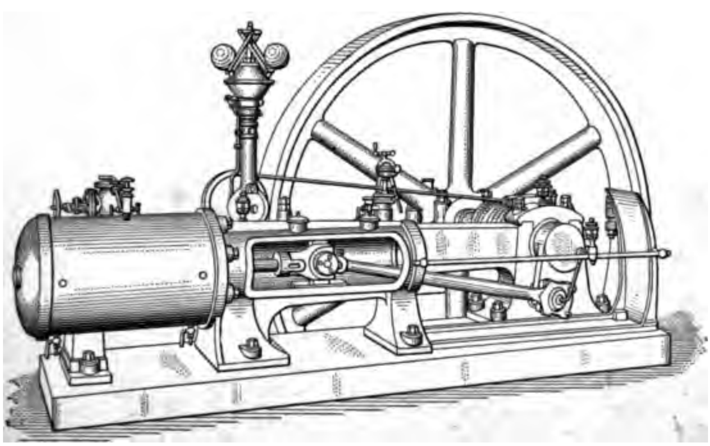


FIG. 4.—Simple steam engine.

These conditions correspond to those of ordinary practice with both Corliss and shaft-governor high-speed engines.

Essential Parts of a Reciprocating Steam Engine.—The crank and connecting rod are used almost universally for converting reciprocating into rotary motion.

The piston traverse in the cylinder = 2 × the effective length of the crank.

Length of Typical Reciprocating Engine.—Between head end and crankpin the length is made up of:

(*a*) Cylinder = two cranks.
(*b*) Piston rod = two cranks.
(*c*) Connecting rod = six cranks (four to eight).
(*d*) All allowances for stuffing-box, cylinder heads, metal in piston, crosshead, etc.

Total = something over ten cranks.

The oscillating engine, having no connecting rod, is only about four cranks long.

The trunk engine, having no piston rod, is about five cranks long.

Engines[1] Classified by Position of Cylinder Axis.—(*a*) *Horizontal;* (*b*) *Vertical;* (*c*) *Inclined.*—Horizontal engines are most usual and cheapest where room or floor space is not limited.

Horizontal Engines.—

Advantages.—

1. Cheapness.
2. Convenience of access from ground level to all parts.
3. Weight distributed over large area for support.

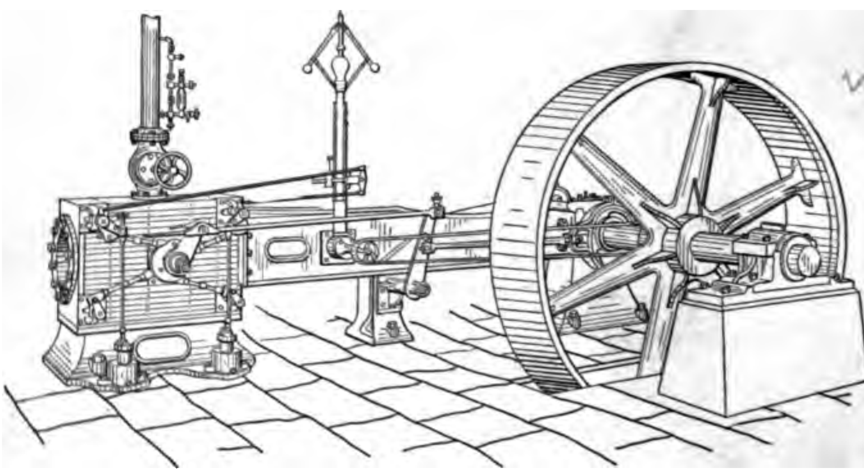


FIG. 5.—Simple Corliss engine.

Disadvantages.—

1. Action of gravity adds to friction. Bad for stuffing-boxes. Piston springs often necessary.
2. Tendency of cylinder to wear oval.

Vertical Engines.—

Advantages.—

1. Diminished ground area.
2. Avoidance of cylinder friction and unequal wear.
3. Require very little cylinder oil.

The small ground-area requirement has made vertical engines prac-

[1] Much of the descriptive material relating to steam engines is from "The Mechanical Engineering of Power Plants," by F. R. HUTTON, John Wiley & Sons, 1908.

tically universal for screw-propelled ships, which are deep-water vessels, and in crowded power plants in cities where ground is costly.

Disadvantages.—

1. Effort on crankpin is greater when weight of mechanism is acting downward with gravity. This must be counteracted to prevent unequal effort on crankpin and irregular speed. Three methods for accomplishing this are used: (*a*) counterweighting the crank on the side opposite the reciprocating parts; (*b*) steam cylinders so calculated as to balance; (*c*) by steam distribution to the two ends of the cylinder.

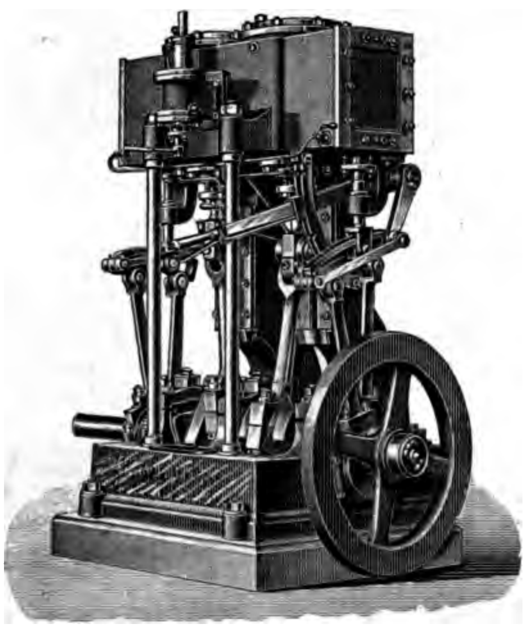


FIG. 6.—Cross-compound vertical reversing engine.

2. In large engines the different parts are on different levels, or stories, increasing the number of men required to handle or superintend them.

Beam Engines.—

Advantages.—

1. Steam cylinder can be vertical.
2. Cylinder and its weight can be kept low down and shaft may also be directly attached to bedplate near the foundation.
3. Long stroke for piston is possible and yet not too much space in ground plan consumed. Great advantage in side-

wheel practice and pumping. R.p.m. may be kept low but piston speed high.

4. Flexibility in alignment of cylinder axis in relation to shaft axis.

5. Where there are several working cylinders, the beam makes easy means of operating them. Some cylinders vertical, others inclined, etc.

Disadvantages.—

1. Too many joints.

2. Weight of beam so far above the center of gravity of the hull.

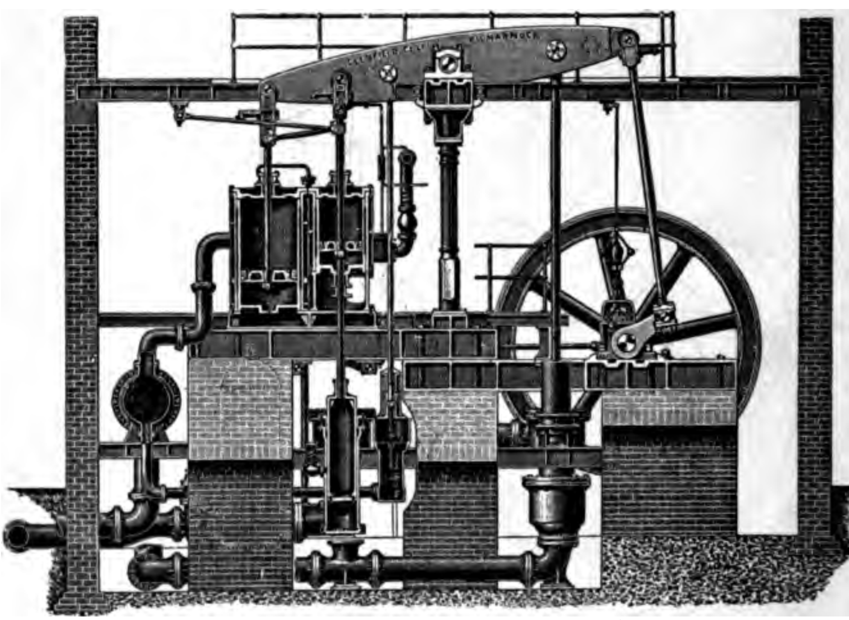


FIG. 7.—Compound beam pumping engine.

3. In warships, vulnerable part exposed. Destruction of this part fatal. This consideration resulted in the back-acting beam type.

Classicatfiion of Engines by Their Use of Steam.—

A. High-speed, low-speed, and moderate speed of rotation.
B. Single- and double-acting.
C. Expansive and non-expansive.
D. Condensing and non-condensing.
E. Simple, compound or multiple expansion.

High-speed Engines.—Consequences of high rotative speed are:

1. Small cylinder volume.
2. Item 1 means engine light in weight.
3. Short length of cylinder means a small crankarm, short connecting rod, and an engine short in length.
4. Variations of either effort or resistance are more promptly met and less noticeable as compared with mean effort or resistance of any given minute.
5. Regulating mechanism tends to equalize effort and resistance in less interval of time than with slower types.
6. Decrease in economy with use, as valves cannot be kept tight.

Low-speed Engines.—Limitations of speed are often imposed by the resistance to be overcome. This condition is met in pumping engines, blowing engines, paddle-wheel engine, marine engines, etc. It is possible to secure a large product of $L \times N$ by making L large when N is small. Engines not making over 125, r.p.m. are classed as low-speed.

Advantages of Low-speed are the Disadvantages of High.—

1. Rapid alternating of admission and compression of steam through ports to cylinder of high-speed engine compel large port areas.
2. Rapid motion of piston compels generous clearance allowance at each end between piston and cylinder heads.
3. Wear per unit of surface greater in short stroke.
4. Heating and abrasive wear goes on rapidly, resulting in possible increase in expense for maintenance and repairs in high-speed engines.
5. Lubrication compelled to be generous to the point of wastefulness in high-speed engines.
6. The above five conditions compel a standard of workmanship in fitting, alignment, provisions for wear, etc., which make high-speed engines costly to build and successful only when well made.

Piston Speed as Distinguished from Rotative Speed.—

Piston speed in feet per minute = $L \times N$.
Less than 500 ft. per minute = low speed.
600 to 800 ft. per minute = moderate speed.
Above 900 ft. per minute = high speed.

As may readily be understood a low rotative speed engine does not necessarily have a low piston speed, as for example, an 18 by 48 Corliss engine making 85 r.p.m. Piston speeds as high as 1,400 ft. have been used.

Single- and Double-acting Engine.—

The single-acting engine takes steam on one side of the piston only. In the vertical engine of this type, the steam acts with gravity and in one direction only.

This results in:

(*a*) Silent running at high speeds.

(*b*) Less danger of overheated bearings.

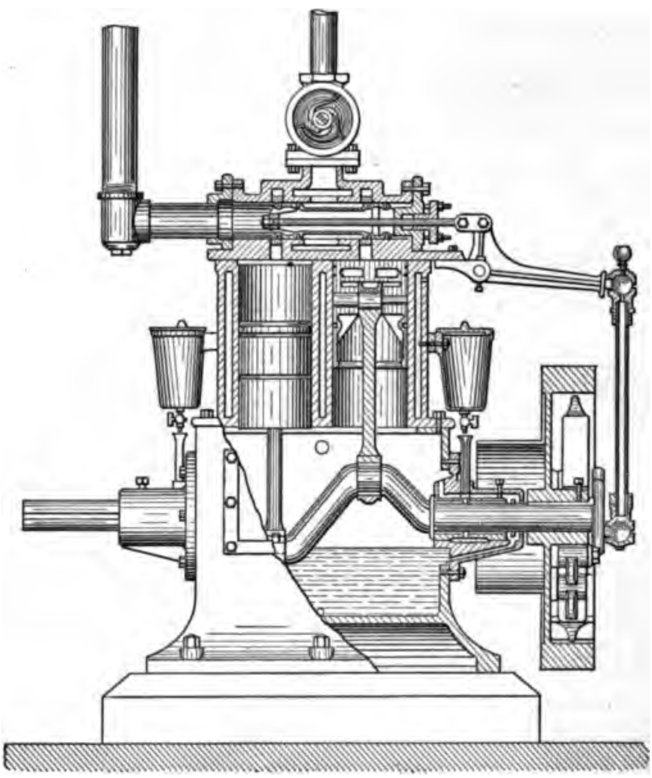


Fig. 8.—Westinghouse single-acting engine.

Single-acting vertical steam engines are usually made with twin cylinders. This construction gives a simple inexpensive engine.

Although a few such types are still on the market, reduction of space occupied per horsepower of output and greater uniformity of turning moment on the crankpin have led to the general adoption of the double-acting principle.

Expansive and Non-expansive Engines.—If steam is allowed to enter the cylinder at full boiler pressure during the entire stroke and is, at the end of the stroke, exhausted at this same pressure, the effort upon the piston has been constant and the steam has been used in the cylinder non-expansively.

If, however, advantage is taken of the elastic quality of steam, it may be admitted to the cylinder at full boiler pressure for a portion of the stroke only and then allowed to expand during the remainder of the stroke.

Under these conditions, the effort upon the piston decreases from the moment the steam supply to the cylinder is shut off until the end of the stroke.

Single-cylinder direct-acting pumps and many elevator engines use steam non-expansively, but the majority of power-plant engines take advantage of the greater economy secured by operating expansively.

Thermal Efficiencies.—The increase in the theoretical thermal efficiency by operating expansively and by condensing is readily seen by an examination of the following figures.

If T_1 = absolute temperature of steam entering the cylinder.
T_2 = absolute temperature of steam leaving the cylinder.

Then

$$\text{efficiency} = \frac{T_1 - T_2}{T_1} = \text{efficiency of ``ideal'' or Carnot cycle.}$$

STEAM-ENGINE EFFICIENCY

Gage pressure, lb.	Absolute pressure, lb.	°F. = t	°F. abs. = T
−13	2	126	586
1	16	216	676
100	115	338	798
114	129	347	807
150	165	366	826
200	215	388	848

$$\frac{798 - 676}{798} = \frac{122}{798} = 15.3 \text{ per cent. efficiency between 115 and 16 lb. abs.}$$

$$\frac{807 - 676}{807} = \frac{131}{807} = 16.3 \text{ per cent. efficiency between 129 and 16 lb. abs.}$$

$$\frac{798 - 586}{798} = \frac{212}{798} = 26.6 \text{ per cent. efficiency between 115 and 2 lb. abs.}$$

$$\frac{826 - 676}{826} = \frac{150}{826} = 18.2 \text{ per cent. efficiency between 165 and 16 lb. abs.}$$

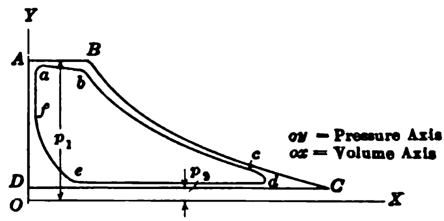
$$\frac{848 - 676}{848} = \frac{172}{848} = 20.3 \text{ per cent. efficiency between 215 and 16 lb. abs.}$$

$$\frac{826 - 586}{826} = \frac{240}{826} = 29.0 \text{ per cent. efficiency between 165 and 2 lb. abs.}$$

The steam engine cannot approach this "ideal" or Carnot cycle. On this account it is customary to compare the thermal efficiency of the actual engine with a modified cycle known as the Rankine cycle with complete expansion.

The outline of this comparison is as follows:

Actual cycle = *abcdefa*.
where ab = steam admission.
bc = expansion.
c = point of release.
de = exhaust stroke.
ef = compression.



p_1 = Steam Line Pressure
p_2 = Condensor or Exhaust Line Pressure

FIG. 9.

Rankine cycle with complete expansion (also called Clausius cycle) = *ABCD*.

Where AB = steam admission.
BC = complete adiabatic expansion down to condenser pressure,
CD = exhaust stroke,
DA = a constant-volume pressure rise.

The thermal efficiency of the Rankine cycle (E_R) is the ratio of the heat changed into work per pound of steam if expanded adiabatically ($H_1 - H_2$) to the heat necessary to raise feed water from the temperature of exhaust to the temperature in the boiler and evaporate it ($H_1 - q_2$),

or $E_R = \dfrac{H_1 - H_2}{H_1 - q_2}$; where

H_1 = total heat per pound of steam at pressure p_1.
H_2 = total heat per pound of steam at pressure p_2 after adiabatic expansion from pressure p_1.
q_2 = heat of the liquid at pressure p_2.

The thermal efficiency of an actual steam engine (E_A) may be expressed as the ratio of the heat actually delivered as work per pound of steam to the heat supplied per pound, measured above the heat of the liquid at the exhaust pressure. The heat equivalent of 1 hp.-hr. is 2,545 B.t.u. Then $\frac{2,545}{W}$ is the heat equivalent of the useful work obtained per pound of steam, W being the pounds of steam supplied per horsepower-hour. Therefore, the thermal efficiency $= \frac{2,545}{W(H_1 - q_2)}$.

Efficiency ratio is a term expressing the ratio between the thermal efficiency of the actual engine and the thermal efficiency of the ideal engine operating on the Rankine cycle with complete expansion between the same pressure limits. Its value will then be the ratio of E_A to E_R, or

$$\begin{aligned}\text{Efficiency ratio } &= \frac{E_A}{E_R}\\ &= \frac{2,545}{W(H_1 - q_2)} \div \frac{H_1 - H_2}{H_1 - q_2}\\ &= \frac{2,545}{W(H_1 - H_2)}\end{aligned}$$

Example.—Determine (*a*) the thermal efficiency, (*b*) the Rankine cycle efficiency, and (*c*) the efficiency ratio of a condensing engine operating with an economy of 13 lb. of dry saturated steam per horsepower-hour, initial pressure 140 lb. absolute, exhaust pressure 2 lb. absolute.

(*a*) $H_1 = 1,192.2$ (from steam tables).
$q_2 = 94.0$ (from steam tables).

$$\text{Thermal efficiency} = \frac{2,545}{13(1,192.2 - 94)} = \frac{195.8}{1,098.2} = 0.178 = 17.8 \text{ per cent.}$$

(*b*) $H_1 = 1,192.2$ (from steam tables).
$H_2 = 914$ (by use of total heat−entropy or "Mollier" diagram).
$q_2 = 94$ (from steam tables).

Efficiency of Rankine cycle $= \frac{1,192.2 - 914}{1,192.2 - 94} = \frac{278.2}{1,098.2} = 0.257 = 25.7$ per cent.

(*c*) Efficiency ratio $= \frac{2,545}{13(1,192.2 - 914)} = \frac{195.8}{278.2} = 0.695$

or from above $= \frac{(a)}{(b)} = \frac{0.178}{0.257} = 0.695.$

Condensing and Non-condensing Engines.—
Advantages of the Condensing Type.

1. With cylinder of given area, stroke, and piston speed the net effective pressure is greater than in non-condensing engines.

Fig. 10.—24 and 38 × 48 cross-compound engine Allis-Chalmers Co.

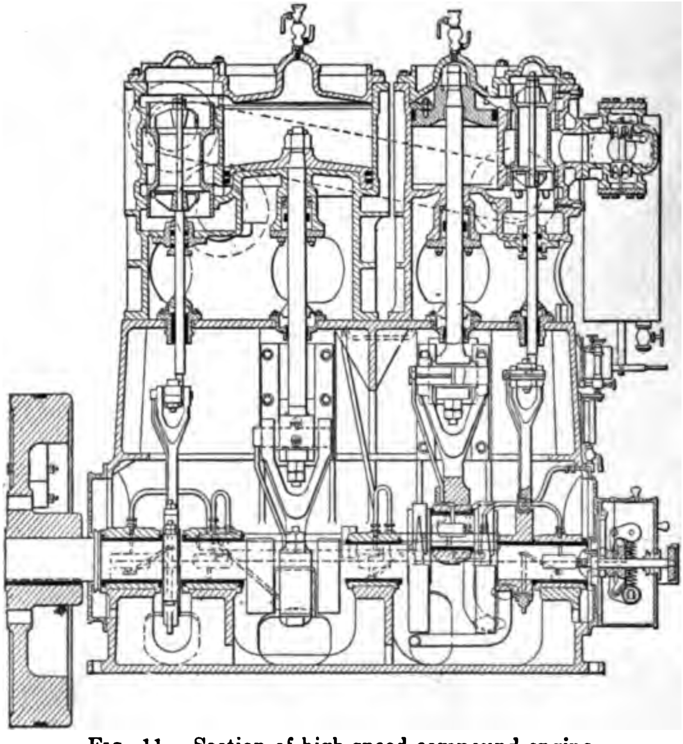


Fig. 11.—Section of high-speed compound engine.

2. Another way of putting it is, same power can be secured by a smaller cylinder with the condensing type.

3. Less volume of steam drawn from boiler per stroke, therefore less coal required per horsepower-hour.

4. Due to more complete expansion the condensing engine utilizes the heat imparted to the steam by the fuel more perfectly than the non-condensing.

5. Efficiency. T_2 might be brought to about 60°F., the ordinary temperature of the cooling water, but it is not often convenient to use so much water, or the cost is prohibitive;

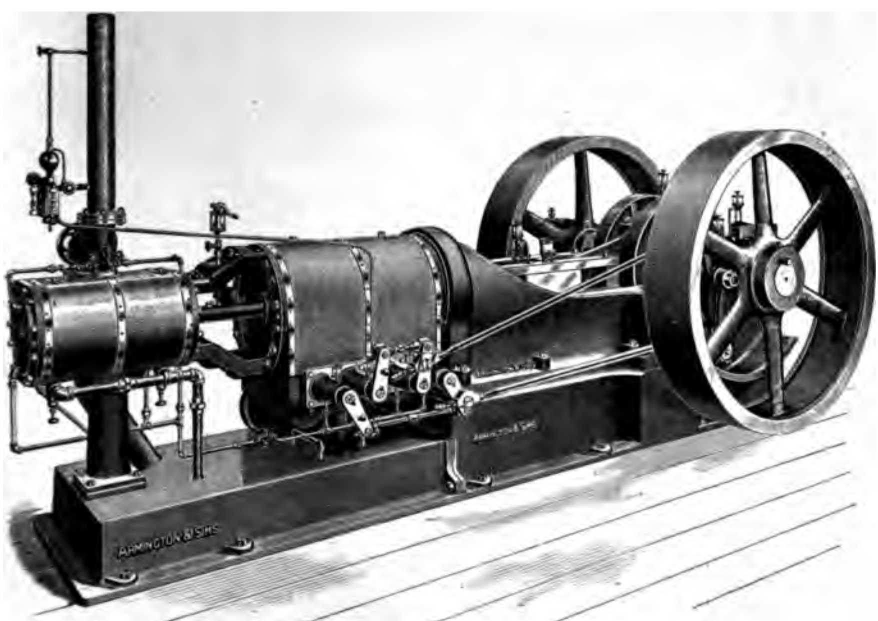


FIG. 12.—Armington & Sims tandem compound high-speed engine.

consequently the temperature is usually about 100° to 130°F. In non-condensing plant, T_2 will be 212°F. or over. Hence the efficiency for the condensing engine is greater.

6. Condensing engine preheats the water to be fed to boiler. This saves fuel and is of advantage to the boiler.

Disadvantages of the Condensing Type.—

1. Low final temperature increases condensation in cylinder thereby reducing economy.

2. Vacuum must be maintained. Engine must do work to accomplish this. Usually circulating water must also be handled.

3. Oil in condensed steam troublesome. Often has to be removed to prevent boiler troubles and clogging of passages.

4. Cannot be used where circulating water is expensive.

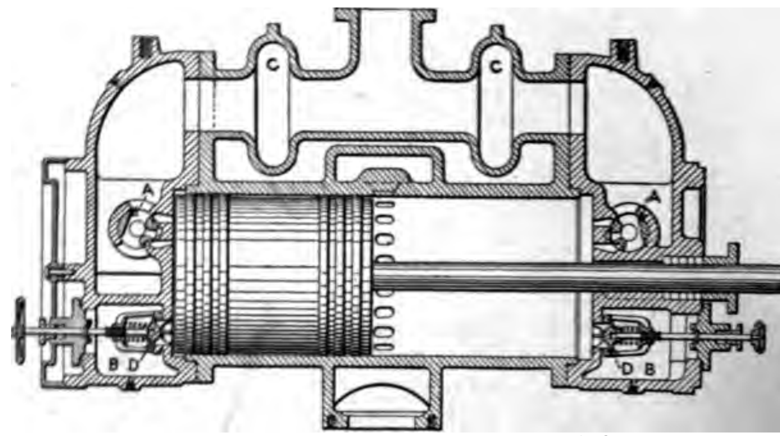


FIG. 13.—Section of una-flow engine cylinder.

Una-flow Engines.—An attempt has been made to combine the advantages of the single-acting engine with those of the double-acting engine by Professor Stumpf, whose una-flow engine is a first-class example of good theory coupled with clever design. The cylinder of this engine is practically twice as long as the ordinary engine cylinder, and the depth of the piston is the stroke of the piston minus the width of the

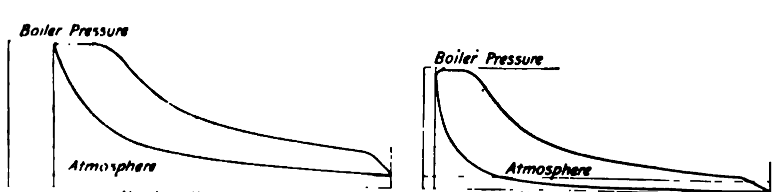


FIG. 14. Indicator cards from non-condensing and condensing una-flow engines.

exhaust ports which are located circumferentially around the center line of the travel of the piston. The admission valves are double-beat poppet valves, located in the cylinder heads. By this construction many of the disadvantages of the double-acting engine are overcome and the good results of the single-acting cylinder are also obtained. Another result of this construction, the ability to carry out the expansion very much further is an advantage of this design. It is possible to expand the steam as fully and as economically in one of these cylinders as in the two cylinders of the ordinary compound engine.[1] As the steam upon exhausting does not come into contact with the admission ports relatively high-cylinder wall temperatures are maintained at the admission ends of the cylinder, thus materially reducing cylinder condensation. High superheats and high vacuums can be readily taken care of and guarantees as low as 8.8 lb. of steam per indicated horsepower-hour have been made by German builders.

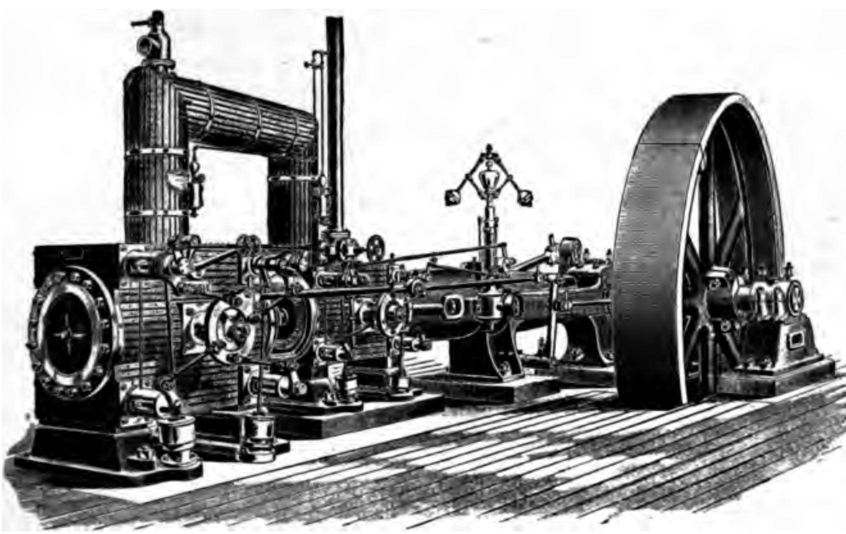


FIG. 15.—Tandem compound Corliss engine.

Compound and Multiple-expansion Engines.—

Advantages.—

1. High expansion and greater difference between initial and final temperature in steam is secured with admission through a longer portion of stroke. Also more favorable crank angles.
2. Greater expansion means higher possible boiler pressure.
3. Strain on mechanism less by receiving high pressure on smaller piston area.
4. More advantageous arrangement for admitting and cutting off steam.
5. Any leakage past valves in high-pressure cylinder goes to low-pressure cylinder and not to waste.
6. Condensation in high-pressure cylinder evaporates and does work in low.

[1] Full details and tests of this construction may be found in PROF. STUMPF's book, "The Una-flow Steam Engine," published by the D. Van Nostrand Co., New York.

7. When so arranged that the several engines have independent crankpins there is an advantage both in size of pin and in crank effort.

8. With cranks quartering or at the proper angles turning effort is equalized thus diminishing weight of flywheel.

9. With reheater the quality of steam may be improved during expansion.

10. Hottest steam in smallest cylinder, thus reducing loss.

11. Range of temperature between initial and final states of each cylinder is less than it would be if expansion were in one cylinder only.

Disadvantages.—

1. Cost of cylinders, other than low.
2. Additional weight and bulk.
3. Friction loss of extra cylinder and valve-chest.
4. Difficulties in governing.
5. Danger of water in low-pressure cylinder, especially troublesome in locomotives.

Throttling and Cut-off Engines.—

Advantages of Throttling.—

1. Engine cheap to build and buy.
2. Steam pressure exerted through considerable portion of stroke, hence less inequality in steam effort at beginning and end of stroke.
3. Throttling effect has a tendency to dry out moisture in steam and to diminish moisture in cylinder.

Disadvantages of Throttling.

1. Not as sensitive as cut-off engine to instantaneous variation in the resistance.
2. Does not regulate as closely to speed as cut-off type.
3. Exhaust usually at higher pressure than in cut-off, causing rejection of more heat.

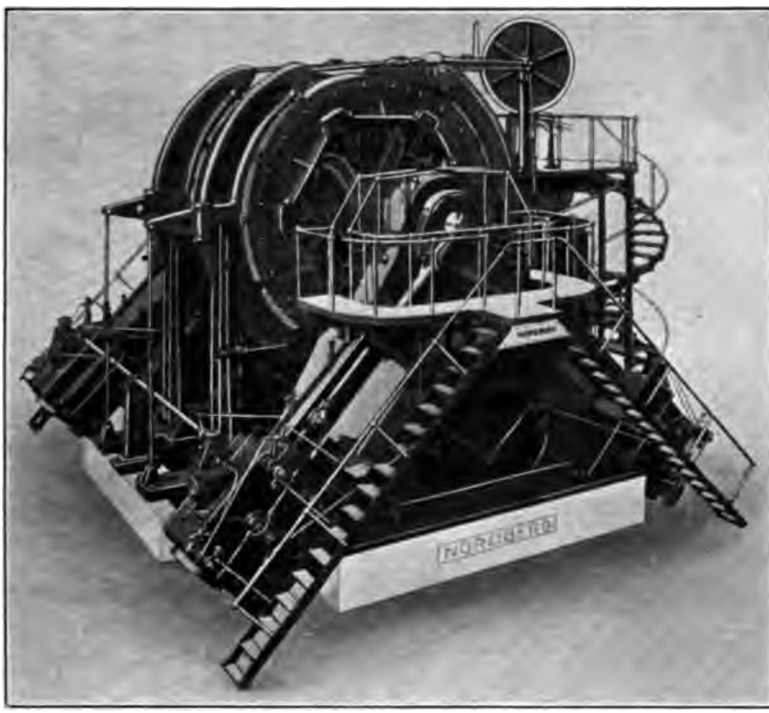
Advantages of Cut-off Engines.

1. Effort controlled per stroke of engine.
2. Engine responds immediately to variations in resistance.
3. More certain to be kept at uniform speed by governor.
4. Full boiler pressure carried on piston until cut-off.
5. Full advantage from expansive working.

Disadvantages.—

1. Wide difference of effort at two ends of stroke requiring massive flywheel.
2. Design and complication of valve-gear.
3. Engine costly to build and buy.
4. Cylinder condensation increased by lower terminal pressure and temperatures.

For many classes of work in power-house service variations are so ride that automatic cut-off is essential. Where effort is constant, as



ꟳIG. 16.—Nordberg four-cylinder steam hoisting engine. Calumet & Arizona Mining Co.

n pumping, in railway and in marine practice, throttling is close enough, specially when the engine driver has to be in constant attendance. The utomatic cut-off is usually more economical, and the engine is usually etter built. When desirable to cut off later than one-third stroke there s little gain in carrying boiler pressure much higher than 80 lb. gage. For simple engines the steam pressure is seldom above 80 lb. gage, but for compound it ranges from 80 to 250 lb.

FIG. 17.—Manhattan type duplex cross-compound engine. Subway Power House, New York.

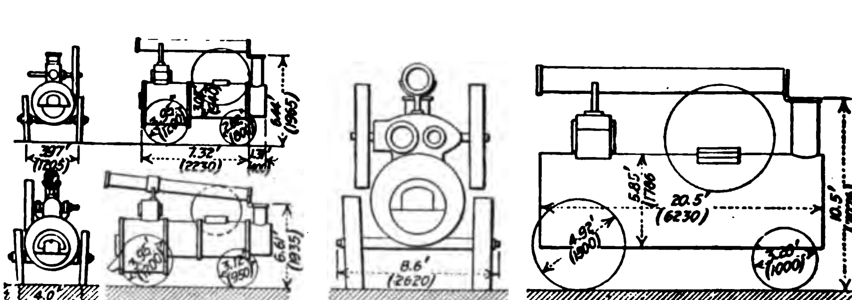


FIG. 18.—Size and types of portable engines.

Triple- and quadruple-expansion engines are used little save for pumping and for marine work. The steam pressure for these engines usually runs from 125 to 250 lb.

Special Classification.—1*st. Stationary*; 2*d Traction;* 3*d Marine.*— The first is subdivided into:

(*a*) **Factory** or mill, including power-house;
(*b*) **Pumping** engines, including blowing engines and air compressors;
(*c*) **Hoisting** engines;
(*d*) **Locomobiles**;
(*e*) **Miscellaneous** engines.

The second is subdivided into:

(*a*) **Locomotives**, traction engines, including road-rollers and self-propelled steam fire engines, auto trucks and automobiles and agricultural engines.

The third consists of engines for marine service.

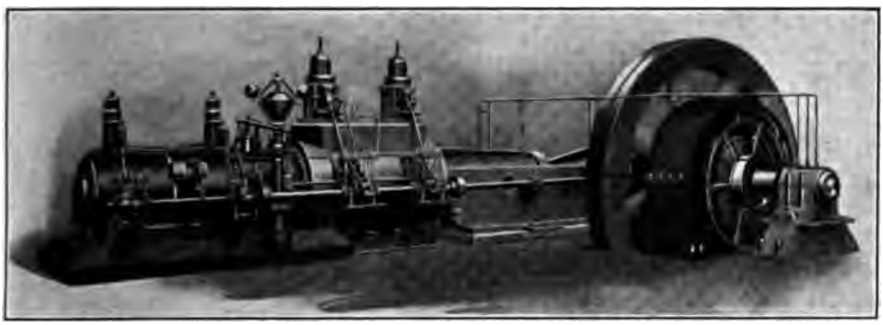


Fig. 19.—Nordberg poppet-valve engine, tandem compound.

Rotary Steam Engines.—

Advantages.—

1. Effort of steam applied directly to produce rotary motion.
2. No reciprocating parts, therefore no inertia effects.
3. No dead centers.
4. Absence of reciprocating parts makes it easy to run at high speed.
5. Very compact. Occupies little room.
6. Either no valve gearing, or very simple if any.
7. Cheap to build. Should be cheap to buy.
8. No reciprocating rods or dead centers, hence condensed steam in cylinder does no harm.
9. Increased construction and item 8 adapt it to outdoor service.
10. No skill required to handle it.

Disadvantages.—

1. Difficulty of satisfactorily packing surfaces which do not move through equal spaces in equal times.
2. Expense connected with proper lubrication. If efficiently lubricated they consume an excessive amount of oil.
3. Excessive waste space to be filled with steam each revolution.
4. In simple type, non-expansive. This coupled with items 1 and 2 make it uneconomical.

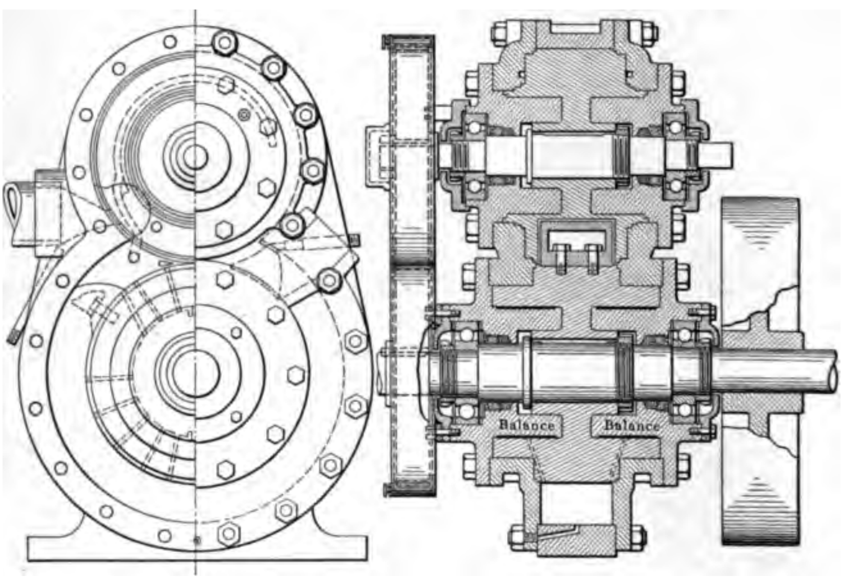


FIG. 20.—Section of Herrick rotary engine.

5. Difficult to design for large horsepowers. Structure becomes inconvenient the moment large areas are desired in order to make $P \times A$ large. Difficult to secure high-piston speed in feet per minute without making the engine excessively large.

Economy may be secured by arranging in series upon a shaft, so that the steam rejected from No. 1 drives No. 2 of larger volume. Few if any rotary engines have been commercially successful. In view of this fact it may be well to record the general data for a rotary engine tested by one of the authors.

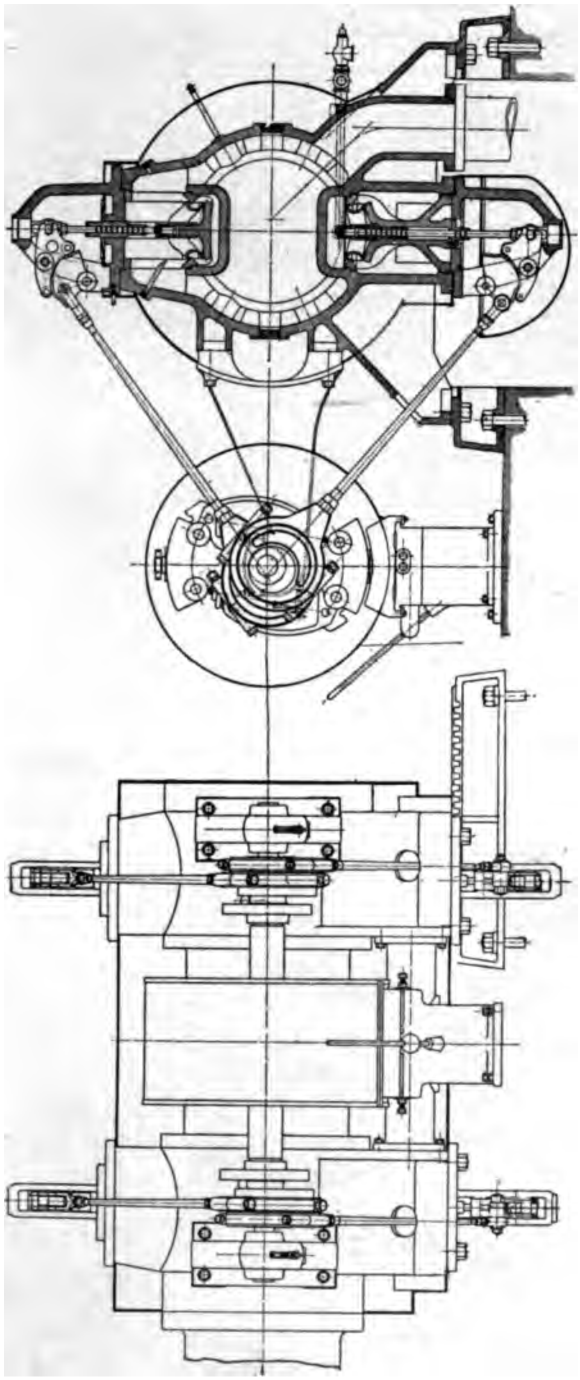


Fig. 21.—Section of poppet-valve engine cylinder.

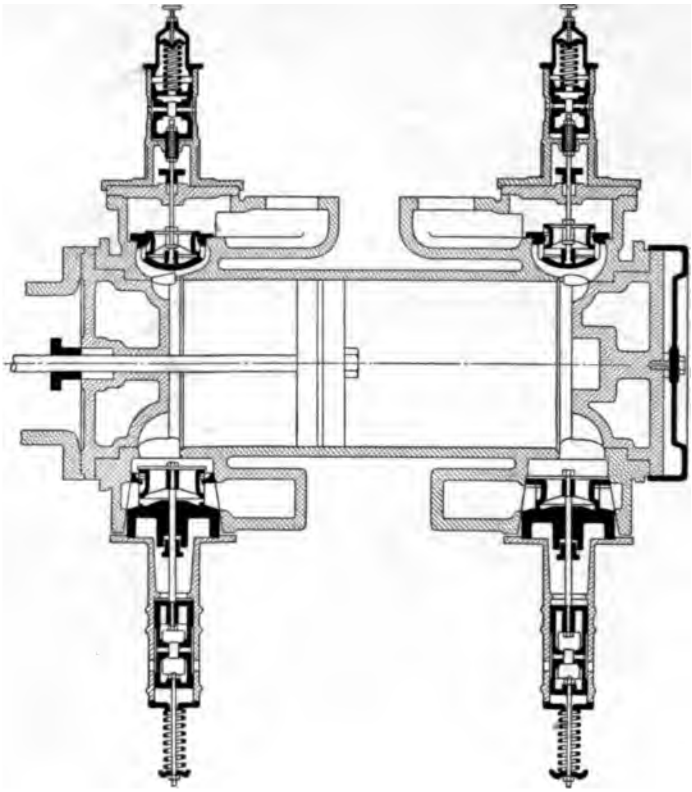


FIG. 22.—Section of Nordberg poppet-valve engine cylinder.

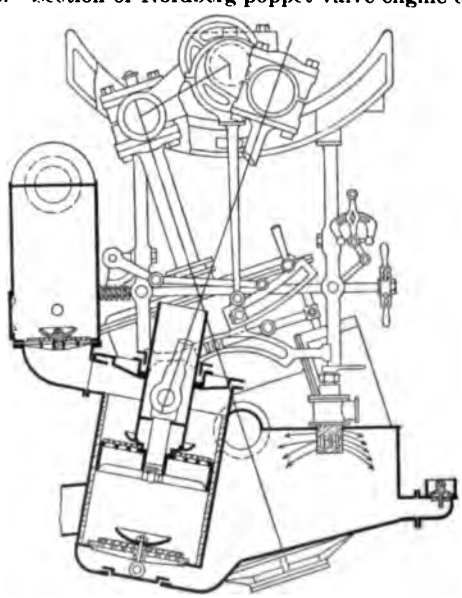


FIG. 23.—Oscillating marine steam engine, section through air pump.

1. The simple steam motor of 20-b.hp. rating occupied only approximately 12 cu. ft. of space overall.

2. The motor which was under load for 5 hr. continuously showed no indications of heating or variations in uniformity of action. Its speed regulation for varying loads was remarkable.

3. The steam consumption of this unit was exceptional, clearly surpassing the corresponding consumption of the average reciprocating unit of similar capacity.

4. After one year of service this rotary engine showed an increased steam consumption per brake horsepower-hour of only 4.6 per cent.

STEAM TURBINES

Basic Principles.—The steam turbine, like the water turbine, utilizes the kinetic energy of fluid in motion. Whenever a moving fluid impinges on moving vanes which change the direction of flow and reduce the velocity of the fluid, the energy of the fluid is converted into mechanical work and is available through the shaft on which the moving vanes are placed.

Differences between Steam and Water Turbines.—There are two important distinctions between steam and water turbines. First, provision must be made in the steam turbine for converting the heat energy of the steam into kinetic energy or the energy of motion. To accomplish this the steam turbine is furnished with nozzles so designed as to control the expansion of the steam in a way to augment its velocity. These nozzles are of two types, diverging and converging. Where the drop of pressure is large the diverging nozzle is used. In this nozzle the walls diverge in the direction of the flow of the steam, so that its outlet area is larger than its inlet area. Where the drop of pressure is smaller the converging nozzle is used. These nozzles differ from the nozzle of the water turbine in that they perform two functions, they not only direct the flow of steam, but they assist in the necessary expansion required to convert the heat energy into kinetic energy. In the Parsons type the fixed blades form a series of nozzles. Second, although jet velocities higher than 300 ft. per second are common on the Pacific Coast, in water turbines the ordinary water velocities are well below this figure. In the steam turbine the velocities are very much greater and the turbine must be adapted to velocities as high as 3,000 to 4,000 ft. per second. It is interesting to compare this speed with the muzzle velocity of a modern rifle ball, which leaves the barrel at about 2,600 ft. per second, or in the neighborhood of 30 miles per minute. This is the speed of steam discharging into the atmosphere from a nozzle of the best shape under a pressure of 50 lb. gage.

In the successful steam turbine:

1. As much of the heat energy of the steam as possible must be converted into kinetic energy.

2. The rotor, nozzles and guide passages must be capable of utilizing the kinetic energy of the steam in the most efficient manner.

3. The casing, rotor and blading must hold their form under the heat strains and centrifugal strains and must be tight against leakage.

4. The apparatus must run at the proper speed at the point of delivery of power and all parts must run within safe limits.

Comparison with the Steam Engine.—In the steam turbine the process of expansion of the steam as in the steam engine is duplicated, except that the flow of steam is continuous instead of intermittent. The steam engine may be termed a ratchet mechanism, while the steam turbine is a continuous mechanism. The difference in form of the turbine and engine is due to the fact that the turbine is designed to work by changing the direction of motion of the flowing steam, while the engine is designed to operate by the direct pressure of the steam. The turbine is thus a velocity motor and the steam engine a pressure motor.

Impulse and Reaction Turbines.—In an impulse turbine the wheel is moved by the impulse of a jet of steam impinging on the blade surfaces. In the true reaction turbine the jet of steam issues from the moving part and impinges on the atmosphere or a fixed blade, thus moving the rotor by reaction. These terms are not used in this way at the present time and the distinction usually made is that in the impulse wheel the expansion of the steam is complete within the nozzle, while in the reaction wheel the expansion is not completed until after the steam enters the moving bucket. These terms are not good terms to use to distinguish the different types of turbines. Another way of stating this difference is that impulse turbines are partial-entry or ventilated turbines, while reaction turbines are full-entry or drowned turbines. It is better not to use these terms.

Classification of Turbines.—Turbines may be classified first as to size into small turbines and large turbines, the small turbine being built in sizes up to 750 hp. or 500 kw., the large turbine, commencing at this size and going up to the limit of mechanical construction, which at the present time may be from 30,000 to 60,000 kw. or larger. Large turbines are usually classified by the names of their inventors or manufacturers and the following types may be distinguished: Parsons, Curtis, Rateau, Zoelly and composite. All of these turbines are very much alike in principle, but differ widely in mechanical design and construction.

The Parsons turbine was the first of the large turbines to be successful and is now manufactured in both Europe and America. The construction is of the drum type in which the blades are fixed in grooves on the outside of a cylindrical drum for the rotor, the fixed blades being held in grooves on the inside of the casing. All Parsons turbines are full intake machines and require for complete expansion from 200 lb. steam pressure down to 28 in. of vacuum, about 80 rows of blades, 40 fixed and 40 moving. No glands are necessary to prevent leakage between the stages, as the pressure differences are quite small, and the clearances at the end of the blades very small indeed. Each manufacturer of Parsons turbines varies the design in minor details, such as thrust bearing, location of dummies, type of blading and mechanical construction of the drum and casing.

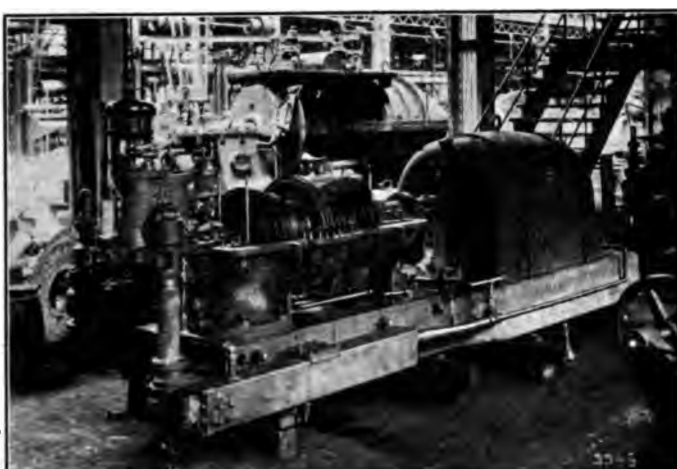


FIG. 24.—Westinghouse double flow turbine on erecting floor.

The Rateau turbine is of the partial-entrance type and so-called multi-cellular construction. The drum system of construction is rarely used, the guide blades are held in diaphragms with glands to prevent leakage where the shaft passes through them. About 20 stages are usually used for the range of expansion between 200 lb. pressure and 28-in. vacuum.

The Zoelly turbine is of the full intake type, but closely follows the Rateau construction in general lines with the exception that more cast iron and cast steel are used in its construction and considerably less riveted-steel work. In this turbine it is very rare that more than 12 stages are used for the expansion of steam from 200 lb. to 28-in. vacuum.

The Curtis turbine, which originated in America, has been classed as the multiple-velocity stage type. It is always partial intake and from three to six stages are necessary for the expansion from 200 lb. to 28-in. vacuum. The particular feature of this type is that in each stage two or more velocity stages may be used. The construction is somewhat similar to the Zoelly machine, with the exception that the shaft has been placed in a vertical position and held by a step bearing. The clearances in this machine can be very generous, as they are in the Zoelly and Rateau types. The diaphragms between the stages are provided with a gland where the shaft passes through them, thus preventing leakage from stage to stage.

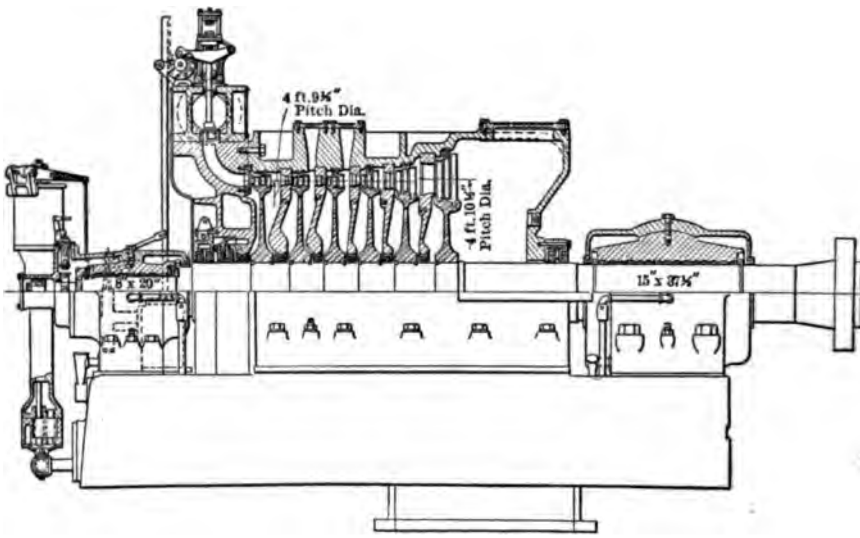


FIG. 25.—7000-kw., 1800-r.p.m., Curtis steam turbine.

The Parsons, Zoelly, and Rateau constructions seem to give much better results in that part of the expansion between atmosphere and 28 in. of vacuum. The Curtis and Rateau types appear to give a trifle better result in the part of the expansion between 200 lb. and atmosphere. When these facts became known some manufacturers started building what we have termed the composite type of machine by using a Curtis wheel for the first stage and the Parsons, Zoelly, or Rateau machine for the low-pressure end. This construction resulted in a shorter, stiffer and cheaper machine and the economies obtained were extremely good. Shortening the machine and decreasing the number of stages enabled the manufacturers to build machines of a size much larger than the old construction, and the simplified wheel constructions enabled higher speeds to be used with the attending economies. Any combination of the various type machines may be made and at the present time practically every turbine manufacturer is turning out the composite machine as the bulk of his product, although machines of the straight Parsons, Zoelly and Curtis type are being built.

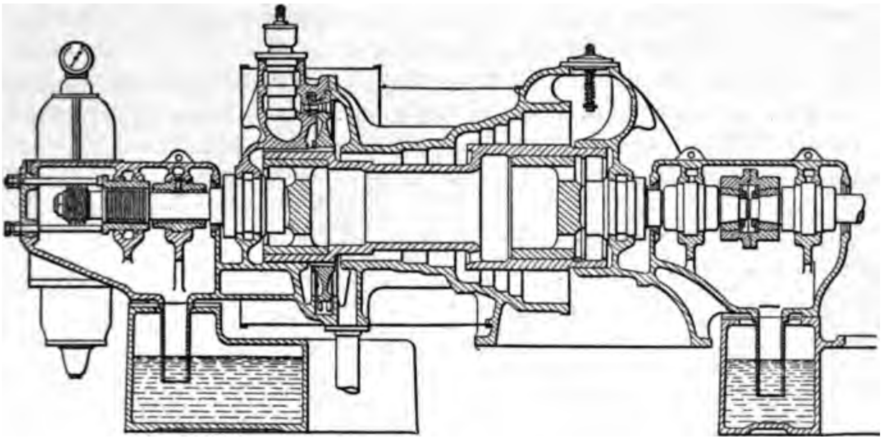


FIG. 26.—Brown, Bouverie & Co. composite-type turbine.

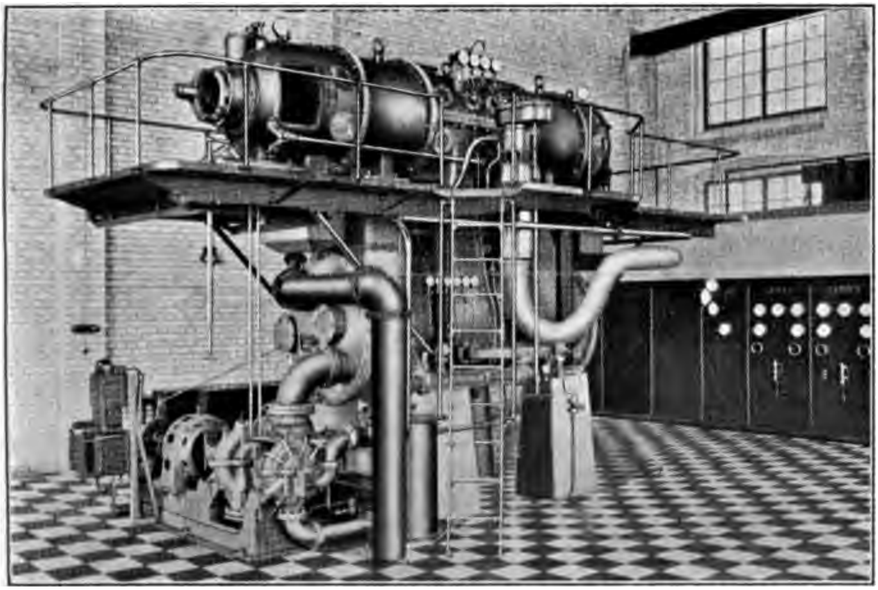


FIG. 27.—1400 kw. Ljungström turbine and condenser, Sandriken, Sweden.

It should be noted that the governing of all full intake machines is of the throttling or puff type, while the governing of the partial intake machines is almost entirely of the nozzle type, that is the steam is admitted at full pressure to one or more nozzles, depending on the load on the turbine.

Quite recently in Sweden the Ljungstrom turbine has been placed on the market, which differs materially from the other types of large turbines. The Ljungstrom turbine is a radial-flow machine, in which the steam is admitted through the center of the shaft and passes through the blades in a radial direction to the condenser. There are no fixed blades, but two sets of working blades moving in opposite directions. By this means bucket speeds may be kept high with reasonable low shaft revolutions. Tests of this machine are extremely good and if

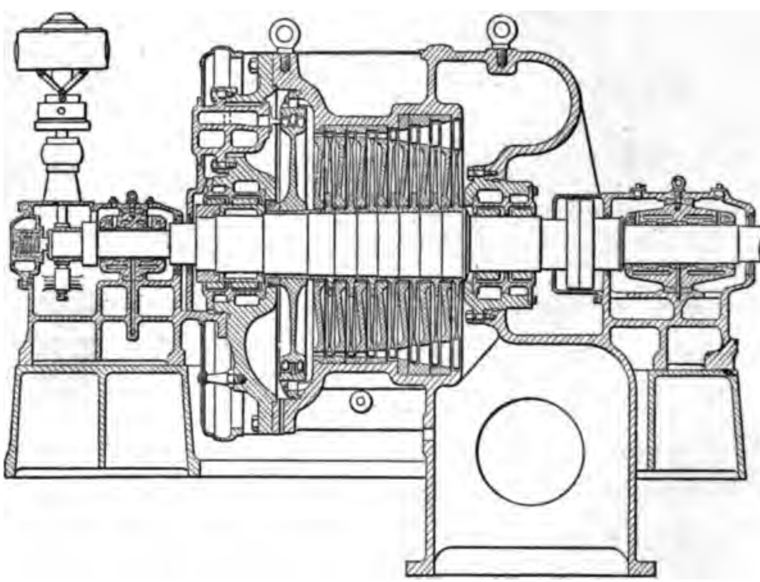


FIG. 28.—Bergmann (Curtis-Rateau) composite type turbine.

the blade construction, which at first sight appears very flimsy, bears the test of time, it may be classed as a fifth-basic type of turbine.

Small Turbines.—Small turbines are of many types, but may be classified by the method in which the steam is used in the wheel. In the DeLaval the steam is expanded in a nozzle and is passed once through the buckets of a single wheel. This was the first successful turbine and has been used to a great extent. This naturally leads to high bucket speeds and in the small sizes to a very high speed of rotation, in some cases as high as 12,000 revolutions per minute. In order to make the turbine a usable proposition a special reduction gearing was made for reducing this speed to the proper point.

In the small turbine of the Curtis manufacture the wheel is provided with two or more sets of blades and the steam is used a number of times on the same wheel, each velocity stage using a portion of the jet velocity.

In the Riedler-Stumpf turbine the nozzle has a number of return passages behind it returning the steam to the wheel buckets.

In the Terry type the nozzle carries behind it a number of ventilated-return passages, by which the steam after passing through the wheel is returned two or three times to the wheel buckets on the entrance side.

In the Electra or Westinghouse type, the steam having passed once through the buckets of the wheel, is caught by a return passage, which returns the steam to the wheel on the discharge side and passes it through the buckets in a contrary direction.

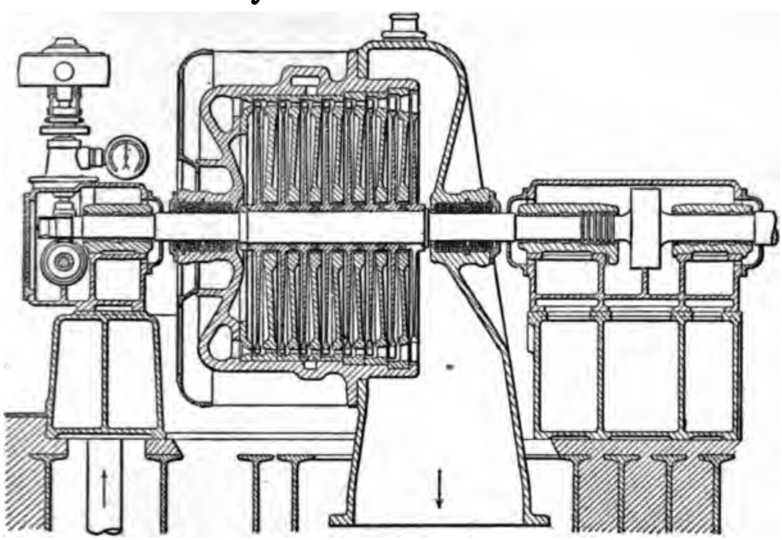


Fig. 29.—Zoelly steam turbine.

In the Kerr type the buckets are almost exactly similar to those of the Pelton waterwheel and the steam is used only once in a set of buckets. This necessitates a number of stages when economy is to be secured.

Most of these turbines, when built in the larger sizes, have more than one wheel and as they increase in size, approach in construction some one of the large turbine types.

By far the largest use for the small steam turbine is for auxiliary work and they are largely run non-condensing. In this case the economy is not quite so good as that of a good steam engine when kept in good condition, the difference being that with the engine the economy will not hold up, whereas with the turbine there is no falling off of steam economy with age and wear. The larger sizes are almost always run condensing, giving economies not quite so good as those of a first-class steam engine under the same conditions.

Rating of Steam Turbines.—Turbine ratings are usually based on maximum sustained load. Momentary overload capacity is very large and moderate overloads of considerable duration can be carried but may require the admission of high-pressure steam to low-pressure stages by means of a secondary valve. Small turbines for driving pumps, blowers, etc., are rated in horsepower. Turbines used to drive electric generators are usually rated in connection with the generator, the combined unit or turbo-generator being rated in kilowatts.

NOTE.—For a review of present turbine construction practice, see paper by PROF. A. G. CHRISTIE, vol. 34, A.S.M.E. *Transactions*, p. 435. Vol. 31 of the same *Transactions* contains a paper by one of the authors which gives sections and steam-consumption curves of most of the types of small turbines.

Efficiency and Losses.[1]—The maximum theoretical efficiency of a steam turbine is the efficiency of the Rankine ideal engine between the temperatures of admission and exhaust.

The several losses which tend to reduce the efficiency of turbines below the theoretical maximum are (1) residual velocity, or the kinetic energy due to the velocity of the steam escaping from the turbine; (2) friction and imperfect expansion in the nozzles; (3) windage, or friction due to rotation of the wheel in steam; (4) friction of the steam traveling through the blades; (5) shocks, impacts, eddies, etc., due to imperfect shape or roughness of blades; (6) leakage around the ends of the blades or through clearance spaces; (7) shaft friction; (8) radiation. The sum of all these losses amounts to about 25 per cent. of the available energy in the largest and best designs and to 50 per cent. or more in small sizes or poor designs.

Oil Required by Steam Turbines.—No cylinder oil is required for the turbine and the exhaust may be condensed and used over and over again as feed water for the boilers without danger, providing exhausts from oily condenser auxiliaries are not permitted to mix with the condensed steam. The only oil required by the turbine is the medium machine oil used for the turbine bearing. This oil is small in quantity, since the bearings of small machines are ring oiled and require to be filled up only about once per month. All large machines have a forced lubrication system for the bearings with a filter and pump attached to the turbine and the oil is used over and over again.

Noise.—The earlier turbo-generators were very noisy, due to the fan action of the rotor. Modern design encloses the stator and the forced ventilation, provided by the fan action of the rotor or an outside blower reduces this noise to a reasonable amount. Where, however, a number of large generators are ventilated from one duct it is well to make provisions for a dampening action and to take particular precautions that

[1] "American Handbook for Electrical Engineers," p. 1399.

an organ-pipe effect is not produced. A great deal of attention has been given to the reduction of noise in turbo-generators, and while it is not possible to entirely eliminate it, the noise has been reduced to a reasonable amount.

Mechanical Efficiency of Steam Engines.—The mechanical efficiency of the same engine will often vary considerably from time to time, depending upon the operating conditions. In general the mechanical efficiency of various types of engines varies from 0.80 to 0.94, although better figures have been secured under test and poorer results are frequently encountered in practice.

The following table gives a few efficiencies secured from tests of engines under normal but good working conditions. The influences of workmanship in the construction of the engine and the variation in operating conditions are apparent from the variation in efficiencies shown.

Kind of engine	Horsepower	Efficiency
Simple engines:		
Horizontal portable	25	0.86
Horizontal portable	80	0.91
High-speed, stationary	50	0.92
High-speed, stationary	100	0.90
Corliss, condensing	150	0.85
Corliss, non-condensing	100	0.86
Compound:		
Portable	80	0.88
Horizontal, stationary	75	0.90
Horizontal, mill engine	300	0.86
Corliss, condensing	100	0.90
High-speed, condensing	46	0.87
Pumping engine	650	0.93
Vertical three-cylinder compound electric-lighting engine	6,000	0.97
Triple-expansion:		
Vertical pumping engines	800	0.94 to 0.97

The following results, secured from three triple-expansion pumping engines under test at St. Louis, indicate the possibilities under conditions of superior workmanship and exceptionally refined conditions of operation.

No. 1		No. 2		No. 3	
I.hp.	Eff.	I.hp.	Eff.	I.hp.	Eff.
873	96.6	875	96.8	859	97.7

Although, strictly speaking, there is no relation between the horsepower capacity of engines and mechanical efficiency, a small engine often being more efficient than a large one, yet in general it is probably true that considering workmanship as a whole, the number of cheap engines of relatively small size and the greater skill and care usually exercised in the operation of large units, a sufficiently close relation between mechanical efficiency and size may be assumed to warrant the use of the following tables of approximate mechanical efficiencies for reciprocating steam engines.

Mechanical efficiency of reciprocating steam engines, rated load		Mechanical efficiency of steam engines in per cent. of rated load efficiency	
I.hp.	Efficiency, per cent.	Per cent. of rated load	Per cent. of rated load efficiency
5	80.0	100	100.0
25	83.5	90	99.0
50	85.0	80	97.2
200	86.5	70	95.2
400	89.0	60	92.5
500	90.0	50	89.0
1,000	90.8	40	83.4
2,000	92.5	30	74.0
3,000	94.0		

Steam-engine Economy.—F. W. Dean says[1] in speaking of the economy of steam plants:

"It is well known that the economics of such plants are very variable and differ from each other. In some cases great care is taken to have good plants carefully operated, while in others they are neglected and incompetent men are employed."

He further says:

"In the case of engines there are non-condensing and condensing engines and turbines, and they can be simple or compound. In addition there are steam engines that are being devised which are likely to surpass in economy any that are now on the market, and these improved engines are on the verge of being introduced (December, 1914). In addition superheated steam is being introduced. By its use it is easy to save 10 per cent. of coal without any accompanying disadvantages. The advantage of using better condensing apparatus is being realized and such apparatus is being introduced with improved economy."

Average Steam Consumption of Reciprocating Steam Engines.—Although the range of steam consumption per horsepower per hour is wide for engines of a given size and of different types, depending upon the quality of construction and the degree of refinement called for by commercial demands, an idea of the average economy of reciprocating steam engines may be had from the following table.

[1] A.S.M.E. *Transactions*, vol. 36, p. 839.

POUNDS OF DRY STEAM PER INDICATED HORSEPOWER-HOUR OF FULL RATED LOAD

I.hp.	Simple high-speed Non-condensing	Simple high-speed Condensing	Simple low-speed Non-condensing	Simple low-speed Condensing	Compound high-speed Non-condensing	Compound high-speed Condensing	Compound low-speed Non-condensing	Compound low-speed Condensing
10	65.0	50.0						
15	57.0	44.0						
20	52.5	40.0						
25	49.0	38.0						
30	46.5	36.0						
40	42.5	33.0						
50	40.0	30.2						
60	38.0	28.5						
75	35.5	26.2						
100	33.0	23.4	27.0	21.6	29.3	22.5	23.6	20.0
150	30.4	21.5	26.3	21.0	28.6	22.0	23.1	19.5
200	29.5	20.6	25.7	20.5	27.9	21.5	22.7	19.0
250	29.0	20.2	25.2	20.0	27.3	21.0	22.3	18.5
300	28.5	20.0	24.8	19.6	26.6	20.5	21.9	18.1
400			24.1	18.8	25.4	19.5	21.1	17.3
500			23.7	18.3	24.2	18.6	20.4	16.5
600			23.4	17.9	23.3	17.9	19.8	15.8
700			23.2	17.7	22.7	17.5	19.2	15.3
800			23.0	17.6	22.3	17.2	18.7	15.0
900			22.9	17.5	22.1	17.0	18.4	14.7
1,000			22.8	17.4	22.0	16.9	18.2	14.5
1,500								13.8
2,000								13.5
2,500								13.2
5,000								12.5

Probable Gain in Steam Consumption by Condensing.—The following table serves to illustrate the marked gain in steam consumption of condensing engines over non-condensing. The figures given are approximations only.

POUNDS DRY STEAM PER INDICATED HORSEPOWER-HOUR

Type of engine	Non-condensing Probable limits	Non-condensing Assumed for comparison	Condensing Probable limits	Condensing Assumed for comparison	Per cent. gained by condensing
Simple high-speed	65–25	33	50–19	23	30
Simple low-speed	30–22	25	24–17	19	24
Comp. high-speed	30–22	26	24–16	20	23
Comp. low-speed	24–18	21	20–12	17	19
Triple high-speed	27–17	22	23–14	17	23
Triple low-speed			18–11	16	

ECONOMY TESTS OF STEAM ENGINES (MARKS AND DAVIS STEAM TABLES)

No.	Kind of engine	R.p.m.	Cylinder dimensions, inches	Superheat °F. or % dryness	Initial press., lb. sq. in., abs.	Exh. press., lb. sq. in. or ins. Hg. absolute	I.hp.	Steam per I.hp.-hr. at initial conditions	Thermal eff. based on I.hp.-hr., percent.	B.t.u. per I.hp.-hr.	Authority	Reference
	Saturated steam											
	Simple non-condensing											
1	Vertical single-valve	354	2-9.5×9	97	107	17.5	35	32.99	7.9	32,060	BARRUS	Eng. Tests, Barrus, No. 16b.
2	Horizontal single-valve	308	8×12	Dry	97	15.4	32	31.2	8.1	31,300	BARRUS	Eng. Tests, Barrus, No. 23b.
3	Willans triple-exp. run as simple engine	406	14×6	*	121	14.7*	34	26.0	9.7	26,250	WILLANS	*Proc.* Inst. C. E., vol. 93, p. 128.
4	P.R.R. locomotive No. 1499	120	22×28	99%	210	15*	975	23.62	10.78	23,670		Loco. Tests & Exhibits, P.R.R.T. No. 116.
	Simple condensing											
5	Four-valve	166	18×30	*	82	4.5″	213	22.08	10.6	23,960	BARRUS	Eng. Tests, Barrus, No. 17, p. 88
6	Four-valve Corliss	59	32.3×60	*	85	5.8″	554	19.45	12.2	20,920	BARRUS	Eng. Tests, Barrus, No. 28, p. 118
7	Four-valve gridiron valve	60	34.4×60	97.5	97	2.1″	613	18.49	12.6	20,200	BARRUS	Eng. Tests, Barrus, No. 22, p. 103
	Compound non-condensing											
8	Vertical cross-compound, single-valve	293	11.5-18.5×13	*	143	16.5*	152	25.20	10.0	25,360	BARRUS	Eng. Tests, Barrus, No. 42B.
9	Skinner tandem	227	16-27×18	98.8%	144	Atmos.	383	21.36	11.9	21,410	CARPENTER	*Power*, July, 1906.
10	West. marine type—Union Station, P.R.R., Pitts	200	17-27×24	*	165	Atmos.	554	20.01	12.5	20,310	SARGENT and LUNDY	*Power*, August, 1903, p. 427.
11	Willans triple-exp. eng. run as compound	401	10-14×6	*	158	14.5*	40	19.19	13.0	19,510	WILLANS	*Proc.* Inst. C. E., vol. 93, p. 128.
12	P.R.R. locomotive No. 2512, four-cylinder	80	14-24×24	98.7%	226	15*	496	19.1	13.4	18,990		Loco. Tests & Exh., P.R.R.T. No. 502.
	Compound condensing											
13	Four-valve, horizontal	75	28-56×60	Dry	166	3.2″	1,714	13.27	17.4	14,610	BARRUS	Eng. Tests, Barrus, No. 55.
14	59 St. Allis Corliss, hor. vert.	...	2-42-2-86×60	10.6°	189	2.0″	7,341	11.7	18.2	14,000	STOTT and PIGOTT	*Trans.* A.S.M.E., 1910, p. 82.
15	Rice Sargent Corliss, Amer. Sugar Co., Bkl	122	20-40×42	Dry	166	1.74″	627	12.10	19.0	13,380	JACOBUS	*Trans.* A.S.M.E., 1903, p. 1274.
16	N. Y. Edison, Wstghse. marine type, Wat. 1	76	43.5-2-76×60	99.3%	201	2.75″	5,442	12.02	19.2	13,260	WHITHAM	
	Triple											
17	Willans triple, non-condensing	415	7-10-14×6	*	187	14.7″	4,000	18.7	12.2	20,880	WILLANS	*Proc.* Inst. C. E., vol. 93, p. 128.
18	Coates mill engine	76	19-29-46×48	99.0%	175	2.25″	787	11.86	19.3	13,210		*Engr.* (Lon.), Aug. 17, 1894, p. 230.
19	Snow pumping engine	21	29-52-80×60	99.0%	169	3.28″	783	11.49	20.2	12,600	GOSS	*Trans.* A.S.M.E., 1900, p. 793.
20	Chestnut Hill pumping engine	177	30-56-87×66	98.6%	200	1.69″	802	10.479	22.0	11,590		*Eng. News*, Aug. 23, 1900, p. 125.

	Quadruple											
21	Nordberg pumping eng., Wildwood	36½	$\frac{19.5\text{–}49.5\text{–}29\text{–}57.5}{42}$	98.0	214	2.3″	712	12.42	22.6	11,270	CARPENTER	*Sibley Jour. Eng.*, April, 1899, p. 247.
22	Nordberg air compressor	57	$\frac{14.5\text{–}22\text{–}38\text{–}54}{48}$	94.3%	257	2.55″	990	11.92	25.52	10,120	HOOD	*Trans.* A.S.M.E., 1907, vol. 28, p. 705.
	Marine											
									(a)	(b)		
23	S. S. Tartar triple-condensing	70	26–42–69×42	......	158	3.7″	1,087	19.83	9.6	26,540	KENNEDY	Inst. C. E., 1890, p. 224.
24	S. S. Meteor triple-condensing	72	29.4–44–70×48	......	160	5.56″	1,994	14.98	9.9	25,700	KENNEDY	Inst. C. E., May, 1891, p. 235.
25	S. S. Iona triple-condensing	61	21.9–34–57×39	......	157	1.66″	645	13.35	11.8	21,650	KENNEDY	Inst. C. E., April, 1891, p. 200.
	Superheated steam											
	Simple condensing											
26	Stumpf uniflow eng.-Flaasaische and Bradford	120	30.5×39	141°	131	2.99″	562	11.02	19.5	13,050	....	The Uniflow Steam Eng., p. 69.
27	Stumpf uniflow eng.-Ehrhardt & Sehmer	131	25.6×39.4	121°	158	3.7″	380	11.06	19.6	12,950	....	The Uniflow Steam Eng., p. 66.
28	Stumpf uniflow eng.-Flaasaische Fab.	121	25.2×39.4	207°	184	4.2″	496	10.28	20.4	12,470	(Alsace Soc. Steam Boiler Owners)	The Uniflow Steam Eng., p. 60.
29	Burmeister & Wain-Stumpf unif. eng.	175	17.7×19.7	309°	155	1.86″	147	9.48	20.8	12,200	BACHE	The Uniflow Steam Eng., p.63.
	Compound non-condensing											
30	White motor car engine	850	3–6×4.5	316°	441	Atmos.	.....	....	17.9	......	CARPENTER	*Trans.* A.S.M.E., 1907, p. 579.
	Compound condensing											
31	Four-valve, horizontal	77	28–54×60	13.0°	136	3.0″	1,252	14.01	16.3	15,640	BARRUS	Eng. Tests, Barrus, No. 48B.
32	Harris Corliss, New Bedford, Mass.	65	30–55×72	14.6°	138	4.4″	1,592	13.50	17.0	14,900	DENTON	*Trans.* A.S.M.E., 1894, p. 882.
33	Boston Edison, McIntosh & Seymour engine	99	29–60×56	98.4°	178	4.3″	2,215	11.88	18.5	13,740	MARKS	*Trans.* A.S.M.E., 1904, p. 491.
34	Cooper Corliss, Atlantic Mills	80	16–40×48	38°	187	2.0″	567	11.293	19.5	13,000	BARRUS	*Eng. Rec.*, Nov. 8, 1902, p. 436.
35	Rice & Sargent hor. cross, Milbourne mills	102	16–28×42	374.5°	157	3.02″*	420	9.56	20.4	12,450	JACOBUS	*Trans.* A.S.M.E., 1904, vol. 25, 264.
36	Easton and Co., horizontal tandem	140	15–24×28	376°	150	2.0″	239	9.0	21.5	11,840	EWING	*Amer. Elec.*, April, 1903, p. 178.
37	Inv. ver. marine type, Cole Marchant and Morley	101	21–36×36	232°	132	2.6″	334	8.68	23.8	10,100	LONGRIDGE	*Engr.* (Lon.), June 2, 1905, p. 546.
	Locomobiles											
38	Wolff compound, semi-portable	233	5.1–9.5×11	279°	187	3.17″	38	10.0	14.2	17,940	MATHOT	*Power*, July, 1906, p. 396.
39	Wolff compound	237	7.9–15×15.8	266°	216	....	109	....	18.3	13,900	GUTTERMUTH	*Zeit.*, V.D.I., 1908, p. 1591.

British Standard used in calculating heat consumption. * Assumed dry.

(a) = thermal eff. based on B.t.u. per indicated horsepower-hour } marine engines and locomobiles.
(b) = B.t.u. in coal per indicated horsepower-hour............ }

The probable steam consumption of condensing engines of different types with different pressures of steam is given in a set of curves by R. H. Thurston and L. L. Brinsmade, *Transactions* A.S.M.E., 1897, from which curves Kent has derived the following approximate figures.

STEAM PRESSURE, ABSOLUTE

Pounds per square inch	400	300	250	200	150	100	75	50
Ideal engine (Rankine cycle)..	6.95	7.5	7.9	8.45	9.20	10.50	11.40	12.90
Quadruple exp. wastes 20 per cent......................	8.75	9.15	9.75	10.50	11.60	13.00	14.00	15.60
Triple exp. wastes 25 per cent.	9.25	9.95	10.50	11.15	12.30	14.00	15.10	16.70
Compound wastes, 33 per cent.	10.50	11.25	11.80	12.70	13.90	15.60	16.90	18.90
Simple engine wastes, 50 per cent......................	14.00	15.00	15.80	16.80	18.40	20.40	22.70	25.20

These engines are of the usual ratios of expansion. A 1 to 7 compound will be as economical as a 1 to 7 triple or quadruple expansion engine.

It is conservative to say that compound engines may now be built to produce an indicated horsepower on 12.5 lb. of saturated steam per hour. With high degree of superheat the 10-lb. mark has been passed but if results were calculated on the basis of saturated steam the figures would barely reach 10 lb.

From 23 four-valve engines in commercial operation, Barrus reports:

1 falls below 12 lb. per horsepower-hour
3 fall below 13 lb. per horsepower-hour
16 fall below 14 lb. per horsepower-hour
only 3 fall above 14 lb. per horsepower-hour

The table on pages 50 and 51 gives an excellent idea of the relative steam economy of different types of engines under test conditions.

One of the recent developments of the reciprocating steam engine in this country (long used in Europe) is the Lentz compound engine which in many respects resembles the modern horizontal, tandem double-acting gas engine.

TESTS OF A 14½ AND 24¾ BY 27½-IN. ENGINE ARE

Steam pressure, lb., gage	Vacuum, in., Hg.	Superheat, °F.	R.p.m.	I.hp.	Lb. steam per i.hp.-hr. under initial conditions
170	26	0	167	366	12.3
170	26	150	167	366	10.4

Tests have recently been reported by the manufacturers which give the steam consumption of a 115-hp. Buckeye-mobile, running at 248 r.p.m.; steam at 210 lb.; initial superheat, 171°F.; non-condensing, as 13.3 lb. per indicated horsepower-hour. This unit produced an in-

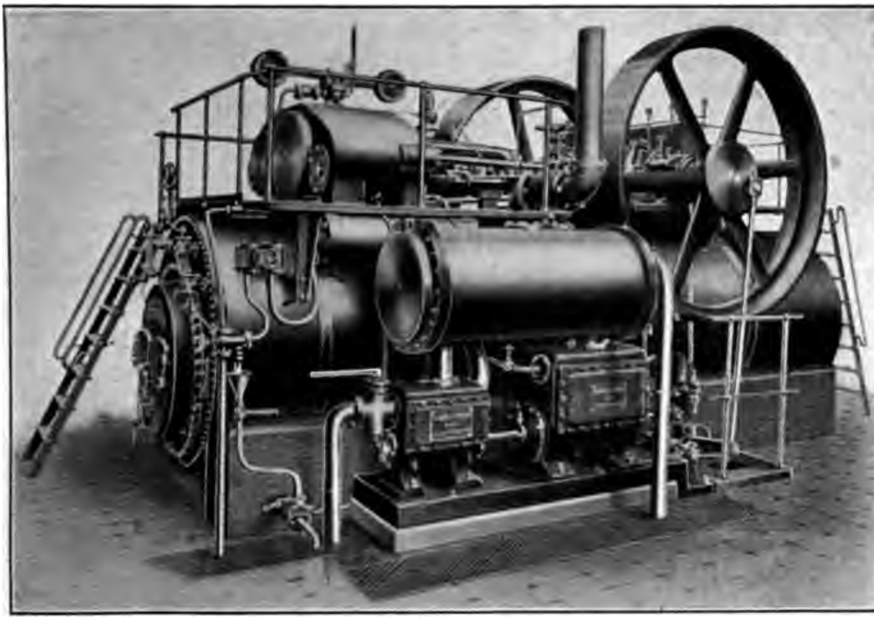


FIG. 30.—Compound locomobile using superheated steam and surface condensers.

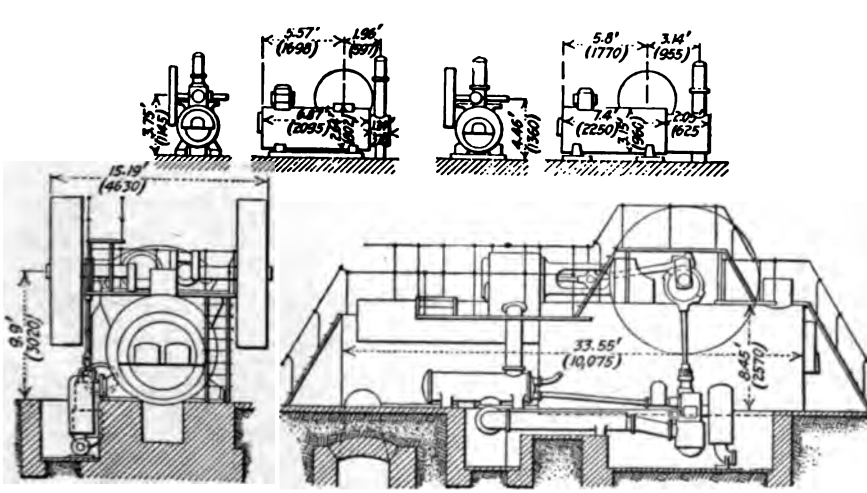


FIG. 31.—Sizes and types of locomobiles.

dicated horsepower-hour on 1.33 lb. of coal having a calorific value of 14,500 B.t.u. per pound. A 169-hp. unit running at 200 r.p.m.; steam at 209 lb.; initial superheat, 218°F.; vacuum 25.7 in., showed a water rate of 9.2 lb. per indicated horsepower-hour. The coal consumption, using fuel with a calorific value of 14,209 B.t.u. per pound, was 1.08 lb. per indicated horsepower-hour.

The effect of superheating upon the steam consumption is brought out more clearly by the following test results. The consumption is given in pounds per horsepower-hour of superheated steam and also in equivalent pounds of saturated steam.

I.hp.	Superheat, °F.	Lb. steam, i.hp.-hr.	Equivalent, sat. steam	B.t.u. per i.hp.-hr.
222	0.0	12.08	12.08	15,000
226	43.7	11.58	11.77	14,600
227	97.7	11.00	11.44	14,200
223	151.7	10.67	11.33	14,000
223	221.2	9.81	10.69	13,300
218	310.9	8.89	10.01	12,000

The result shows that for every 100°F. superheat the steam consumption per indicated horsepower-hour was reduced 1 lb. or 8.5 per cent. and that the consumption expressed in terms of equivalent saturated steam was reduced 0.6 lb. per indicated horsepower-hour. It should be remembered that the poorer the engine the larger the gain from the use of superheat. In a first-class engine the gain in economy is rarely over 5 per cent. per 100°F. superheat, but in a poor simple non-condensing engine it may be 50 per cent. for the first 100° superheat.

Steam Consumption of Small Steam Turbines.—The majority of small steam turbines are run non-condensing and are rated on the brake horsepower instead of the indicated. Although the steam consumption for different-sized turbines may vary with the speed, an estimate of the average consumption for non-condensing units may be secured from the following table.

Pounds of Dry Steam per Brake Horsepower-hour at Full Rated Load

(180 lb. steam pressure, moderate superheat, no back pressure)

B.hp.	Lb. per b.hp.-hr.
10	60
25	50
50	45
100	40
150	35
200	30
250	28

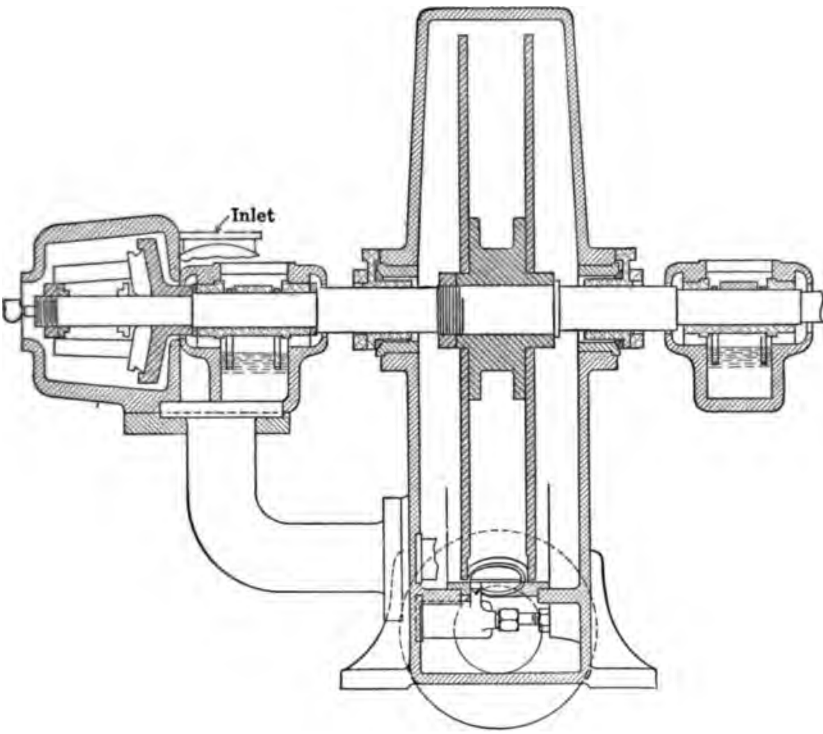


FIG. 32.—Section of Terry steam turbine.

The values would be materially increased by back pressure. Estimates of the probable steam consumption of the larger condensing steam turbines may be made from the following tables, keeping in mind that the figures recorded are test results under the best operating conditions.

RECORDS OF STEAM CONSUMPTION FOR TURBINES

Turbine	Nominal power	Steam used per hr., lb.			Estimated equivalent per i.hp.
		B.hp.	E.hp.	Kw.	
With saturated steam					
Westinghouse Parsons	400 kw.	13.63	.....	.....	12.68
Westinghouse Parsons	1,250 kw.	.....	14.13	18.95	12.72
Rateau	300 hp.	.....	14.90	.....	13.11
Curtis (American)	500 kw.	13.63	.....	.....	12.68
DeLaval	300 hp.	15.17	.....	.....	13.96
Zoelly	500 hp.	.....	16.05	21.50	14.12

Record of Steam Consumption for Turbines (*Continued*)

Turbine	Nominal power	Steam used per hr., lb. B.hp.	E.hp.	Kw.	Estimated equivalent per i.hp.
With superheated steam, not exceeding 150°F.					
Parsons	3,000 kw.	.....	11.79	15.80	10.85
Westinghouse Parsons	400 kw.	12.07	.....	.....	11.23
Curtis (American)	500 kw.	.....	13.28	17.79	11.95
Parsons	1,500 kw.	.....	13.44	18.00	12.10
Zoelly	500 hp.	.....	14.05	18.82	12.36
Westinghouse Parsons	1,250 kw.	.....	13.78	18.48	12.40
DeLaval	300 hp.	13.94	.....	.....	12.82
Parsons	300 kw.	.....	14.96	10.06	13.16
Curtis (English)	500 kw.	.....	15.29	20.50	13.76
With highly superheated steam, superheat from 180° to 290°F.					
Parsons	5,000 kw.	.....	11.00	14.74	10.12
Curtis (American)	500 kw.	.....	11.26	15.10	10.14
Curtis (American)	2,000 kw.	.....	11.27	15.12	10.36
Westinghouse Parsons	400 kw.	11.17	.....	.....	10.39

Economy Tests of Turbines (Marks and Davis Tables)

In calculating Rankine B.t.u. an efficiency of 90 per cent. is assumed for turbine and generator

Turbines	Load, kw.	Steam press. abs.	Super-heat, °F.	Vac. 29.92 in. Bar.	Lb. steam per kw.-hr.	B.t.u. per kw.-min.	Rankine cycle, B.t.u. per kw.-min.	Eff. ratio
Dunstan Parsons	6,257	204.0	176.0	29.02	11.95	249.4	190.8	76.4
A. E. G. Rummelsburg	2,177	198.5	272.0	29.32	11.77	257.1	180.9	70.4
Erste Brunner	6,000	191.0	194.0	28.12	12.58	259.5	205.6	79.2
Chic-Edison Curtis	8,191	199.0	143.0	29.36	12.68	263.9	184.1	69.8
A. E. G. Moabit	3,169	185.0	215.0	29.00	12.70	268.4	192.8	71.8
Carville Parsons	5,164	215.0	121.0	28.96	13.18	268.8	192.4	71.6
Bergman	1,545	195.0	201.0	28.54	12.97	270.3	199.1	73.6
Zoelly Charlottenburg	2,052	200.0	202.0	28.39	13.05	271.6	200.5	73.9
Erste Brunner	2,000	161.5	118.0	27.82	13.84	274.5	218.4	79.6
Zoelly Augsburg Nurnberg	1,250	188.0	204.0	28.79	13.10	274.5	196.2	71.5
Boston Edison Curtis	5,195	189.0	142.0	28.74	13.52	276.1	199.3	72.2
Richmond Allis	4,328	186.0	108.0	27.97	14.02	278.3	211.6	76.0
Brown Boveri	3,500	162.0	133.0	28.80	13.72	278.6	203.3	72.9
City Elec. Westinghouse	8,563	183.0	59.0	28.10	14.43	280.5	211.8	75.5
B. R. T. Westinghouse	11,601	192.0	114.0	27.82	14.23	282.8	212.3	75.1
Manchester Howden	6,383	203.0	137.0	27.40	14.30	285.8	213.7	74.8
N. Y. E. Westinghouse	9,870	192.0	97.0	27.19	15.05	294.6	219.5	74.5
N. Y. E. Curtis No. 10	8,921	190.0	111.0	28.10	14.86	296.0	208.8	70.5
Varberg DeLaval	1,570	172.3	170.5	28.49	16.47	337.9	206.6	61.2

References for Economy Tests of Turbines

Turbine	Reference
Dunstan Parsons	*London Engineering*, Mar. 10, 1911.
A. E. G. Rummelsberg	Stodola, p. 404, 4th edition.
Erste Brunner	*Zeit.* V. D. I., Dec. 10, 1910, p. 2104.
Chicago Edison Curtis	Report on Tests by Breckenridge, 1907.
A. E. G. Moabit	*Zeit.* V. D. I., 1907, p. 386.
Carville Parsons	Stodola, 439, 4th edition.
Bergman	*Zeit.* V. D. I., Dec. 10, 1910, p. 2104.
Zoelly Charlottenburg	Escher Wyss Leaflet.
Erste Brunner	*Zeit.* V. D. I., Dec. 10, 1910, p. 2104.
Zoelly Augsburg Nurnberg	*Zeit.* V. D. I., Dec. 10, 1910, p. 2104.
Boston Edison Curtis	N. E. L. A. *Proceedings*, 1907, p. 433.
Richmond Allis	*Sibley Journal*, January, 1911.
Brown Boveri	*Zeit.* V. D. I., Dec. 10, 1910, p. 2104.
City Electric Westinghouse	A.S.M.E., *Journal*, December, 1910, p. 2089.
B. R. T. Westinghouse	Operating Co.
Manchester Howden	*London Engineer*, 1909, p. 462.
N. Y. E. Westinghouse	N. Y. E. Tests, 1907.
N. Y. E. Curtis	N. Y. E. Tests, Whitham, 1907.
Varberg DeLaval	*Power*, May 3, 1910, p. 798.

F. W. Dean (*Power*, May 6, 1913) gives the following comparative figures for steam consumption for reciprocating steam engines and steam turbines.

Guaranteed Steam Consumption
Pounds per Kilowatt-hour
(150 lb. steam pressure)

Capacities, kw.	Engine sat. steam (Vac. = 26 in.)	100°F. superheat (Vac. = 26 in.)	Turbine 100°F. superheat (vac. = 28 in.)
500	18.67	16.8	17.7
1,000	18.80	16.9	18.0
1,500	18.80	16.9	16.8
2,000	18.93	17.0	16.6

Low-pressure or Exhaust, Bleeder and Mixed-pressure Turbines.—Since that portion of the turbine which utilizes the expansion of the steam from around atmospheric pressure to the best obtainable vacuum is much more efficient than the high-pressure portion of the turbine, it was proposed very early in the turbine development to install low-pressure or exhaust turbines in connection with non-condensing engines already installed. The combination unit gave nearly double the power of the engine unit alone, was somewhat less costly per unit capacity and resulted in a saving of from 5 to 10 per cent. or even more in operating costs. These savings were particularly large in connection with the exhaust from rolling mill, hoisting and reversing engines and steam hammers, but as these machines were of intermittent service some method of providing steam during the period of rest had to be introduced. This led Professor Rateau, one of the earliest to use this type of machine, to invent his regenerator, which provided storage to tide over the intermittent periods of stopping. Other manufacturers got around the difficulty in another way by the introduction of an auxiliary Curtis high-pressure wheel ahead of the low-pressure element and providing a governor valve which admitted high-pressure steam to run the turbines during the periods when no low-pressure steam, or not enough, was available. These machines are known as mixed-pressure turbines and have become quite common, as they are well suited to certain conditions. The Rateau regenerator practically consists of a large cast-iron tank

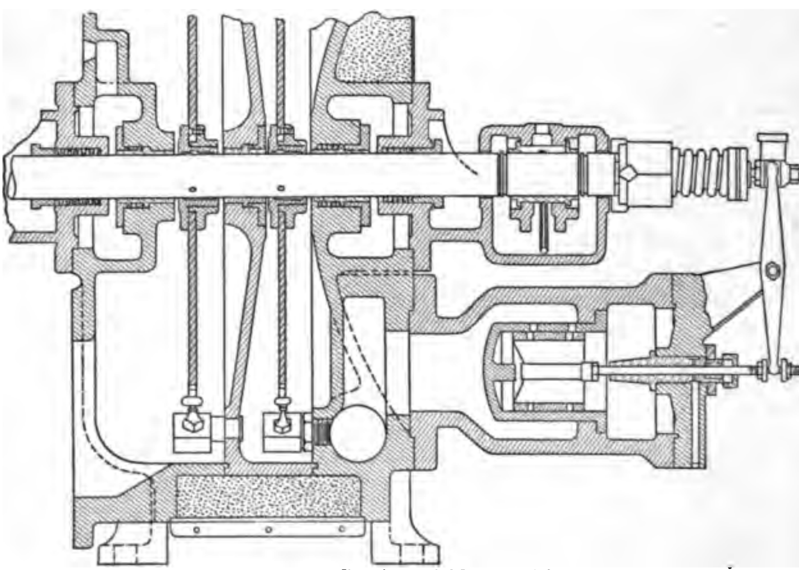


FIG. 33.—Section of Kerr turbine.

acting as a jet condenser at atmospheric pressure. A slight reduction in the pressure vaporizes some of the stored water, thus providing steam for the low-pressure turbine. Combinations of regenerator and mixed-pressure turbines are also used with good success. (For the theory of regenerators, see paper by F. G. Gasche, Engineers' Society of Western Pennsylvania, Nov. 19, 1912.)

It is sometimes convenient, especially in district lighting and heating systems, to abstract steam from the turbine at a little above atmospheric pressure, this steam to be used in the heating system or for similar uses. Such a turbine is known as a bleeder turbine. The steam is taken off through a specially designed valve arranged to maintain any predetermined pressure in the bleeder main.

By combining[1] the superior efficiency of the engine in the pressure

[1] "American Handbook for Electrical Engineers," p. 1395.

range above the atmosphere with that of the turbine below the atmospheric range a resultant superior to the efficiency of either single type may be secured. Standard piston engines are able to sustain full rated load when run non-condensing and often will carry from 25 to 50 per cent. above rated load without danger of excessive wear. Under such conditions the water rate per kilowatt-hour is high but all the heat rejected in the exhaust is available to a low-pressure turbine; hence the net economy of the combined engine and turbine may be considerably superior to that of the engine when run condensing. By proportioning the turbine to efficiently utilize the exhaust steam and by connecting to it a high-vacuum condenser, the initial capacity of the unit may be doubled or even trebled, and if the engine is in good condition, the resultant efficiency may be superior to that obtainable from a new complete-expansion turbine of equal capacity.

Combined Engine and Turbine Unit.—Owners of first-class reciprocating steam-engine plants will often be confronted with the desirability of the extension of their plants. This may be done in a number of ways, by the duplication of the engine units, by the purchase of complete expansion turbines, or by installing low-pressure turbines, to operate on the exhaust steam from the existing steam engines. It is difficult to say which of these ways is the best, as the local condition will largely govern the problem, but it is safe to say that in no case at the present time in plants of 1,000 hp. or larger, will the reciprocating engine plant be duplicated. In a few cases it has been found advisable to install low-pressure turbines, but in most instances complete expansion turbines, replacing the original engines, will be the most economical solution. Each case, however, should be considered by itself, bearing in mind that the cost of the low-pressure steam turbine will be about two-thirds of that of a complete expansion turbine capable of developing the power of the engine and low-pressure turbine combined.

Economy of Combined Engine and Turbine.—An improvement of from 20 to 25 per cent. in steam economy is obtained in combining the low-pressure turbine with a compound condensing engine of normal cylinder proportions, and from 40 to 45 per cent. with the same engine non-condensing.

Considering a single-cylinder Corliss engine in connection with the low-pressure turbine, the customer's coal bill could be decreased from 50 to 60 per cent.

With the use of a low-pressure turbine the excellent performance of large, efficient reciprocating engines can be bettered by about 2.5 lb. or over 14 per cent.

Messrs. Stott and Pigott reported[1] the following results from com-

[1] A.S.M.E. *Journal*, March, 1910.

bining a 7,500-kw. Manhattan-type engine with a 7,500-kw. low-pressure turbine.

(a) An increase of 100 per cent. in maximum capacity of plant.

(b) An increase of 146 per cent. in economic capacity of plant.

(c) A saving of approximately 85 per cent. of the condensed steam for return to the boilers.

(d) An average improvement in economy of 13 per cent. over the best high-pressure turbine results.

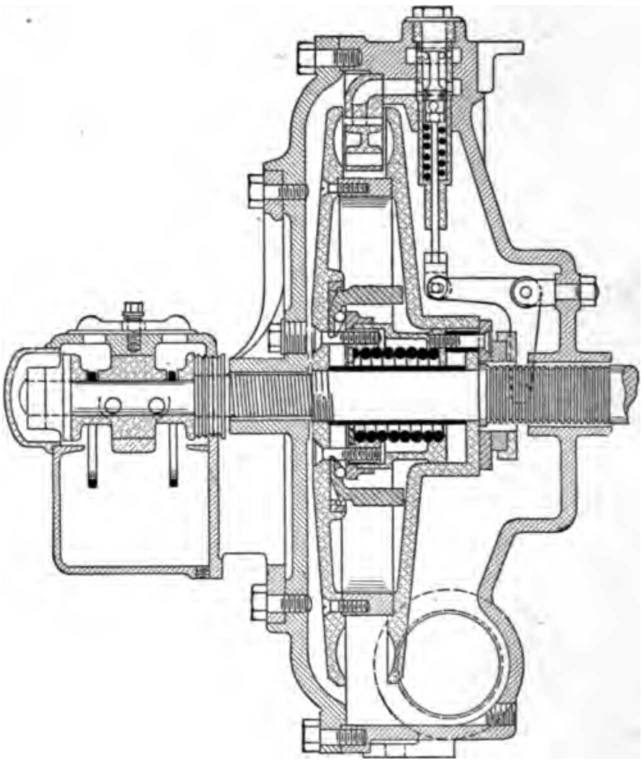


FIG. 34.—Section of Dake-American turbine.

(e) An average improvement in economy of 2.5 per cent. (between the limits of 7,500 and 15,000 kw.) over the results obtained by the engine units alone.

(f) An average thermal efficiency between the limits of 6,500 and 15,500 kw. of 20.6 per cent.

Variable-load Steam Consumption.—The steam consumption given both for reciprocating engines and turbines are for full rated load. For other loads the economy is not as good. The average variation of steam consumption per horsepower-hour or per kilowatt-hour with change of load, expressed in terms of per cent. of full-load economy, may be taken as:

PER CENT. OF FULL LOAD
STEAM CONSUMPTION PER HORSEPOWER-HOUR OR PER KILOWATT-HOUR

Load =	1/4	2/4	3/4	4/4	5/4
Engine	160	120	105	100	103
Turbine (small non-condensing)	135	110	105	100	101
Turbine (large condensing) (185 lb. and 100°F. superheated steam, 28 in. vac.)	114	107	101	100	101

Duty of Pumping Engines.—The duty, efficiency and economy of reciprocating triple-expansion pumping engines are shown by the following table of "Official Trials."

Type	Location	Rated capacity millions of U. S. gallons	Water actually pumped, millions of U. S. gallons 24 hr.	Net head pumped against, lb. per sq. in.	R.p.m.	Initial gage pressure	Indicated horsepower	Developed horsepower	Mechanical efficiency	Dry steam per i.hp.-hour	Duty: Per thousand lb. of dry steam	Duty: Per one million B.t.u. in steam
Holly	Louisville, Ky.	24.0	24.111	90.0	24.0	155.1	925.7	879.4	95.0	9.64[1]	195.0[1]	164.5
Holly	Frankfort, Pa.	20.0	21.219	95.7	20.1	180.2	.....	817.0	.....	.....	184.4	
Holly	Albany, N. Y.	12.0	12.193	139.5	22.3	153.0	.....	726.0	.....	.....	182.1	
Holly	Brockton, Mass.	6.0	6.316	130.6	40.1	150.0	.....	334.0	.....	.....	170.0	
Holly	Cleveland, O	2.5	2.142	180.7	62.3	149.6	158.7	151.9	95.8	11.51	164.6	148.8
Allis	Boston, Mass.	30.0	30.314	61.0	17.7	185.5	801.5	747.8	93.3	10.33	178.5	163.9
Allis	St. Louis, Mo.	20.0	20.070	104.0	16.5	140.6	859.2	839.6	97.7	10.66	181.3	158.8
Allis	St. Louis, Mo.	15.0	15.121	127.0	16.4	126.2	801.6	726.3	90.6	10.67	179.4	158.1
Allis	Milwaukee, Wis.	12.0	12.430	121.0	20.4	124.6	673.0	618.0	91.8	10.82	175.4	151.0

[1] 109°F. superheat at throttle.

Complete details of three such pumping engines are presented.

Make and type	Holly vertical	Holly vertical	Worthington horizontal
Contract price	$124,700	$112,679	$5,500
Weight, tons	970	970	
Capacity, gal., 24 hr	25,000,000	25,000,000	15,000,000
Diameter, cyls., in	32, 60, 90	34, 64, 98	36, 72
Stroke, ft	5	5½	4
Diameter, plungers, in	36	34¾	36½
R.p.m.	21.89	21.356	13.142
Condensers	Jet	Surface	Jet
Steam pressure, lb. gage	150	160	100
Water pressure, lb.	80	108	80
Duty, ft.-lb. per million B.t.u.	151,000,000	160,000,000	
Duty per 1,000 lb. 150° superheated steam	179,000,000	193,500,000	
Duty per 1,000 lb. saturated steam	....	....	125,000,000
I.hp.	966	1,118	
Water hp.	896	1,074	560
Mech. efficiency	94.5	96.3	
Steam per w.hp.-hr., lb.	11.071	10.23[1]	15.8
Steam per i.hp.-hr., lb.[1]	10.250	9.86[1]	

[1] 150°F. superheat.

Cost of Simple, High-speed Engines.—

I.hp.	Cost f.o.b.	Cost per i.hp. f.o.b.	Cost erected	Cost per i.hp. erected
50	$760	$15.20	$910	$18.20
75	910	12.10	1,070	14.30
100	1,090	10.90	1,270	12.70
125	1,260	10.00	1,420	11.30
150	1,410	9.40	1,625	10.80
200	1,735	8.70	1,990	10.00
250	2,050	8.20	2,350	9.40

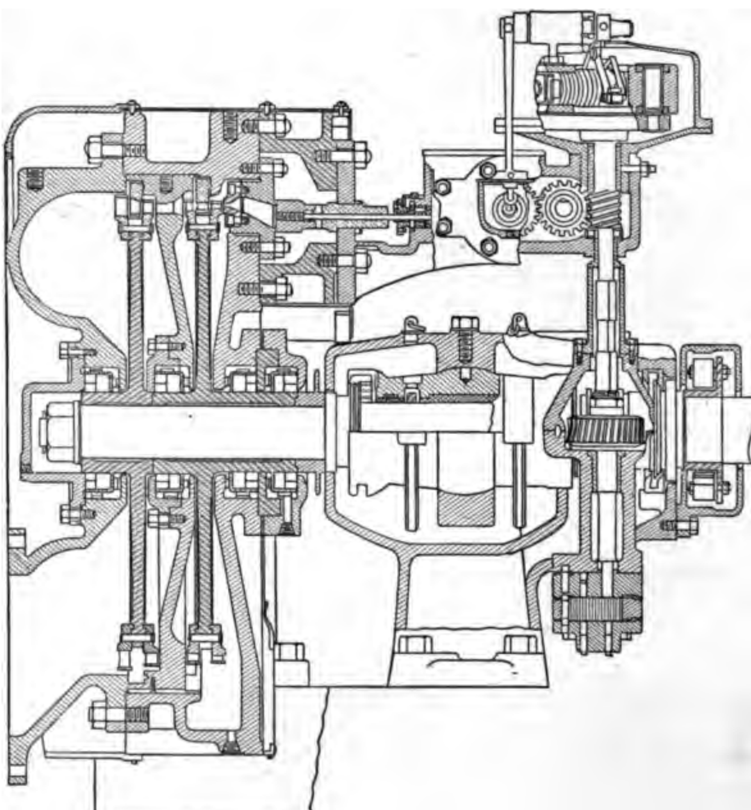


FIG. 35.—Section of 125-kw. two-stage Curtis turbine.

The above figures are averages of several quotations for different makes of engines and serve as a basis for approximate cost estimates.

The f.o.b. cost in dollars for these engines may be represented by the formula

$$435 + 6.5 \times \text{i.hp.}$$

Many such formulæ are presented by different writers. Naturally they vary considerably in accordance with the types of engines considered and the state of the market.

Many authors subdivide into many divisions, but for work of this character this seems unnecessary.

The average of six such formulæ for simple, high-speed engines up to 500 i.hp. is, cost in dollars = 200 + 10 × i.hp.

Above 500 i.hp. the approximate formula seems to approach more nearly to, cost in dollars = 200 + 6 × i.hp.

The erecting cost of such engines is reported by Potter[1] as

Engine, i.hp.	Erecting cost
75	$125 to 150
100	150 to 200
150	200 to 300
300	300 to 400
450	400 to 450
600	400 to 600

The erecting costs indicated for the engines listed on page 62 follow roughly the formula, erecting cost in dollars = 100 + 0.8 × i.hp., which averages about 16.5 per cent. of f.o.b. cost of engine.

Another set of figures for average costs of such engines, erected, including foundations, compiled from the wide experience of one consulting engineer, is as follows:

Engine horsepower.........	10	12	14	15	20	30	40	50	75
Cost per horsepower.........	$36.50	$36.00	$35.50	$35.00	$34.50	$28.50	$21.50	$17.40	$15.50

Cost of Simple Non-condensing Corliss Engines.—The cost of simple, low-speed engines may be obtained from the table below.

It should, however, be borne in mind that the prices given may now be seriously affected by steam-turbine competition. When such is the case, it is probable that the cost of steam engines may be reduced to approximately 70 per cent. of the f.o.b. prices given.

In order to establish the ratio of cylinder size to horsepower of the engine, the dimensions of the various engines are included.

The horsepowers are based on 80 lb. gage pressure and cut-off at ¼ stroke.

[1] *Power*, Dec. 30, 1913.

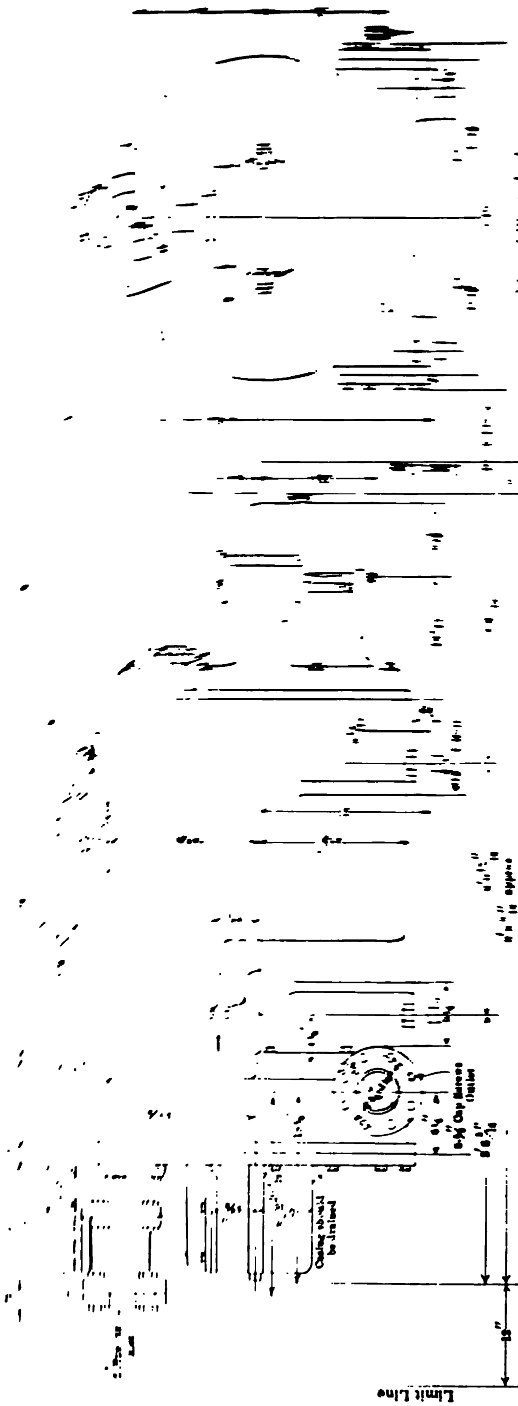


FIG. 30. 35-kw. non-condensing Curtis steam turbine.

COST OF SIMPLE NON-CONDENSING CORLISS ENGINES

Size	Hp.	Engine f.o.b.	Per hp.	Foundation	Erecting	Piping	Total	Total per hp.
14 × 36	100	$1,700	$17.00	$275	$175	$165	$2,315	$23.15
14 × 42	110	1,800	16.40	300	200	175	2,475	22.50
16 × 36	125	1,950	15.60	325	210	180	2,665	21.30
16 × 42	140	2,000	14.30	350	225	190	2,765	19.75
18 × 36	155	2,150	13.90	375	240	200	2,965	19.15
18 × 42	175	2,350	13.40	400	250	210	3,210	18.35
18 × 48	200	2,600	13.00	425	260	220	3,505	17.50
20 × 42	210	2,600	12.40	500	270	230	3,600	17.10
20 × 48	230	2,850	12.35	525	275	250	3,900	16.95
22 × 42	250	3,000	12.00	550	300	310	4,160	16.65
22 × 48	280	3,500	12.50	600	325	340	4,765	17.00
42 × 48	320	4,000	12.50	700	375	390	5,465	17.10
26 × 48	380	4,650	12.20	800	440	560	6,450	16.95
28 × 48	425	5,150	12.10	900	500	800	7,350	17.30
28 × 54	450	5,300	10.80	1,050	575	950	7,875	17.50
30 × 48	500	5,800	11.60	1,200	600	1,070	8,670	17.30

The variation in prices listed is considerable and precludes the application of a positive formula.

The following will, however, approximate the values sufficiently closely for preliminary estimates.

$$\text{Cost in dollars, f.o.b.} = 700 + 10 \times \text{i.hp.}$$

The averages of other formulæ given by different writers for the cost of simple Corliss engines are:

Up to 400 i.hp.

$$820 + 10.3 \times \text{i.hp.}$$

Above 400 i.hp.

$$375 + 10.2 \times \text{i.hp.}$$

which are in reasonable agreement with the formula above.

The erecting cost of these engines, including foundations, seems to be from 35 per cent. to 50 per cent. of the f.o.b. cost of the engines, averaging about 37 per cent. The following formulæ may serve in this connection.

Erecting cost in dollars =

$$\text{(up to 400 i.hp.) } 275 + 3.5 \times \text{i.hp.}$$
$$\text{(above 400 i.hp.) } 250 + 5.0 \times \text{i.hp.}$$

COST OF COMPOUND HIGH-SPEED NON-CONDENSING ENGINES

Size	Hp.[1]	Cost f.o.b.	Cost per hp. f.o.b.
8 and 13 × 12	60	$1,190	$19.85
8 and 16 × 12	80	1,420	17.75
10 and 18 × 12	100	1,520	15.20
11 and 19 × 14	125	1,830	14.65
12 and 20 × 16	150	2,285	15.20
13 and 22 × 16	200	2,620	13.10
15 and 25 × 16	250	2,890	11.55
16 and 28 × 18	300	3,580	11.90
17 and 30 × 18	350	4,150	11.85
20 and 36 × 18	400	4,590	11.50

[1] Hp. based on 100 lb., steam pressure.

Although these quotations vary considerably, they correspond approximately to:

$$\text{Cost in dollars} = 500 + 10.5 \times \text{i.hp.}$$

The averages of several other formulæ for the cost of such engines are:

Cost in dollars =

$$(\text{below 250 i.hp.})\ 775 + 10.5 \times \text{i.hp.}$$
$$(\text{above 250 i.hp.})\ 625 + 9.5 \times \text{i.hp.}$$

With no data at hand relating to the cost of setting compound high-speed engines, it may be assumed that this item amounts to about the same percentage of the initial engine costs as in the high-speed simple engine. This will give a basis for forming approximate estimates.

Cost of Compound Condensing Corliss Engines.—

Size	Hp.	Wt. lb. inc., condenser	Wt. per hp.	Cost[1] f.o.b.	Cost per hp.	Foundation and erecting	Total cost	Total per hp.
Based on 100 lb. steam pressure								
14 × 28 × 42	200	60,000	300	$4,565	$22.80	$1,050	$5,615	$28.10
18 × 34 × 42	300	85,000	283	5,700	19.00	1,025	6,725	22.40
20 × 38 × 48	400	110,000	275	7,300	18.25	1,250	8,550	21.35
22 × 42 × 48	500	140,000	280	8,480	16.95	1,400	9,880	19.75
24 × 46 × 48	600	170,000	284	10,000	18.85	1,675	11,675	19.60
Based on 120 lb. steam pressure								
13 × 26 × 42	200	60,000	300	4,465	22.80	1,050	5,515	27.55
16 × 32 × 42	300	85,000	283	5,500	18.33	1,025	6,525	21.75
18 × 36 × 48	400	110,000	275	7,100	17.75	1,250	8,350	20.85
20 × 40 × 48	500	140,000	280	8,280	16.55	1,400	9,680	19.35
22 × 44 × 48	600	170,000	284	9,900	16.50	1,675	11,575	19.30

[1] Prices include condensers.

The prices given apply to both tandem and cross-compound engines, the cost of the former being less than 10 per cent. lower in smaller sizes and somewhat greater in large sizes.

The corresponding approximate formulæ are:

Cost in dollars, f.o.b. = 1,800 + 13.6 × i.hp.
and cost in dollars, f.o.b. = 1,600 + 13.6 × i.hp.

The average of other formulæ for compound Corliss engines up to 600 hp. is:

Cost in dollars, f.o.b. = 1,500 + 9.8 × i.hp.

The cost of foundations and setting seems to be about 18 per cent. of the f.o.b. cost of the engine, for units of the sizes given.

One firm manufacturing Corliss engines of from 200 to 3,000 hp. gives the weights of engines as from 200 to 250 lb. per horsepower and the price from 6 to 8 cts. per pound.

Cost of Compound Condensing Engines.—Figures reported by one consulting engineer indicate that in general the average cost per horsepower, erected, for various types of compound condensing engines is approximately as follows:

Engine, horsepower......	100	200	300	400	500	600	700	800	900	1,000	1,500	2,000
Cost per horsepower.	$25	$24	$23	$22	$21.50	$21.25	$21	$20.75	$20.50	$20.25	$19.50	$19

Cost of Steam Turbines.—Small turbines for driving pumps, blowers, etc., cost from $20 to $40 per horsepower.

All types of turbines cost approximately the same. Turbine costs are subject to considerable variation, the tendency being a decided decrease in the cost per kilowatt from year to year. The costs given should therefore be checked against actual quotations, even when used in preliminary estimates.

Approximate Cost of Steam Turbines and Generators

In Dollars per Kilowatt, Rated Capacity

Size, kw.	100	300	500	750	1,000	2,500	5,000	7,500	10,000
Direct-current condensing..................	48	38	36	30	28				
60-cycle A.C. condensing..................	...	...	22	...	16	13.50	12.00	11.00	
25-cycle A.C. condensing..................	...	...	...	...	25	17.00	14.00	12.50	10.50

F. W. Dean[1] gives the comparative cost of turbine and reciprocating engine units on the basis of real outputs at 80 per cent. power factor including apparatus and exciters, all erected, including foundations as:

COMPARATIVE COSTS

Capacities, kw.	Horizontal four-valve engine units		Turbine units		Ratio engine to turbine
	Total cost	Cost per kv.a.	Total cost	Cost per kv.a.	
500	$22,700	$45.40	$12,250	$24.50	1.85
1,000	40,200	40.20	17,900	17.90	2.24
1,500	62,200	41.50	23,800	15.85	2.61
2,000	76,400	38.20	30,500	15.25	2.50

Commercial Aspects of the Turbine.—Limitations. The field of the turbine is limited by its relatively high speed and the fact that it is non-reversible. Because of its high speed it cannot be used for driving machinery by belting. In view of this restriction the turbine is limited to driving direct-connected apparatus, such as electric generators, centrifugal pumps, fans and blowers, ship propellers, etc. The turbine is essentially a central station apparatus.

Field of the Reciprocating Engine.—The power for rollingmills, blast furnaces, mine and water-works pumping, mine hoisting, air and ammonia compressors, etc., will be furnished by the piston engine for a long time, and Corliss engines or similar types will still be used for mill work where belt or rope driving is preferred.

Turbine Advantages.—The advantages of the turbine are high economy under variable loads, small floor space, uniform angular velocity and close speed regulation, freedom from vibration, inexpensive foundations, ease in erecting and usually quickness in starting, steam economy not seriously impaired by age, small cost of maintenance and attendance, adapted to use with high superheat, water of condensation free from oil.

Engine Advantages.—Rather than call attention to special features engine builders point to reliability, simple and cheap condensing system requiring only a small quantity of condensing water, and to the fact that nearly as good economy as the best turbine economy may be obtained without the use of highly superheated steam.

It is only fair to say that in the last 5 years all of the builders of large-sized engines for land work have practically gone out of business, although many engine builders still remain in the field. It is noticeable that only the builders of the higher-class engines and the lower-class engines in the medium and small sizes remain in the field.

[1] Paper before National Association of Cotton Mfgrs., Boston, April, 1913.

Turbines vs. Engines in Units of Small Capacity.—Under this title K. S. Barstow presented the following summary before the A.S.M.E. in December, 1915, for units of less than 500-hp. capacity.

APPLICABILITY OF TURBINES

1. Direct-connected units, operating condensing. 60-cycle generators in all sizes, also 25-cycle generators above 1,000-kw. capacity. (This paper is, however, not intended to deal with units of this size.)

Direct-current generators in sizes up to 1,000-kw. capacity, including exciter units of all sizes.

Centrifugal pumping machinery operating under substantially constant head and quantity conditions, and at moderately high head, say from 100 ft. up, depending upon the size of the unit.

Fans and blowers for delivering air at pressures from 1½-in. water column to 30 lb. per square inch.

2. Direct-connected units, operating non-condensing for all the above purposes, in those cases wherein steam economy is not the prime factor or where the exhaust steam can be completely utilized, and, in the latter case, particularly where oil-free exhaust steam is desirable or essential.

3. Geared units, operating either condensing or non-condensing for all the above-mentioned applications, and in addition, many others which would otherwise fall in the category of the steam engine, on account of the relatively slow speed of the apparatus to be driven.

APPLICABILITY OF ENGINES

1. Non-condensing units, direct-connected or belted and used for driving:

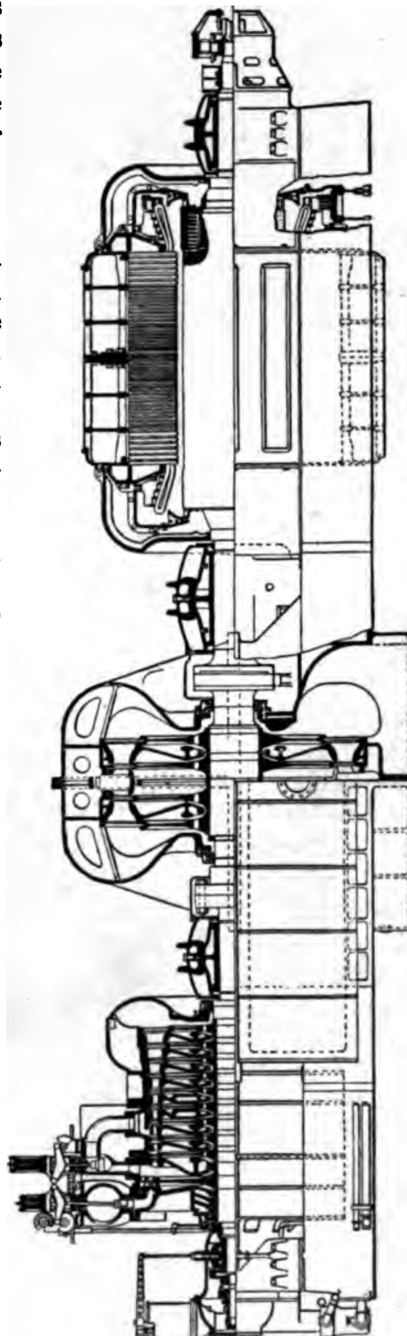


FIG. 37.—35000-kw. General Electric Co. turbine.

Electric generators of all classes excepting exciter sets of small capacity, unless belted from the main engine. Centrifugal pumping machinery, operating under variable head and quality conditions and at relatively low heads, say up to 100 ft., depending on the capacity of the unit. Pumps and compressors for delivering water or gases in relatively small quantities and at relatively high pressures in the case of pumps at pressures above 100 lb. per square inch and in the case of compressors at pressures from 1 lb. per square inch and above.

Fans and blowers (including induced draft fans) for handling air in variable quantities and at relatively low pressures, say not over 5-in. water column.

Line shafts of mills, where the driven apparatus is closely grouped and the load factor is good.

All apparatus requiring reversal in direction of rotation, as in hoisting engines and engines for traction purposes.

2. Condensing units direct-connected or belted, for all the above purposes, particularly where the condensing water supply is limited, and where the water must be recooled and recirculated.

The Saving of Space.—The introduction of the composite type of machine has made possible large improvements in the space requirements of turbines. At the present time 60,000 kw. in turbines with their condensing apparatus can be placed in the space occupied by 8,000 kw. of vertical engines with direct-connected generators and jet-condensing apparatus. A few years ago, three 20,000-kw. machines were placed in the space formerly occupied by four 4,500-kw. engine units. A few years earlier an 8,000-kw. turbine was placed in the space occupied by a 4,000-kw. vertical engine. Where horizontal engines have been in use the space saving is of course much larger and 60,000 kw. could be placed today with its condensing apparatus in the space occupied by a 1,500-kw. duplex tandem compound horizontal unit. In the smaller sizes the saving of space is nearly as well marked, but is always a function of the speed of the turbine, slow-speed turbines being very large and high-speed turbines comparatively very small. A 250-hp. medium-speed non-condensing turbine can be put inside of a 4-ft. cube, and a 150-hp. high-speed turbine might be put in a 30-in. cube. There seems to have been no great saving in space with the vertical-shaft turbine over the horizontal-shaft turbine when condensers and auxiliaries are taken into account.

Based on the overall dimensions of the generating units but not including condensers and auxiliaries, W. F. Fisher reports (*Power*, vol. 34, page 275) the average approximate floor space per engine horsepower to be:

COMPARATIVE SPACE PER E.HP.

Capacities, hp.	Turbine units		Corliss engines units	
	Type	Sq. ft. per e.hp.	Type	Sq. ft. per e.hp.
2,000	Westinghouse-Parsons	0.146	Horiz. cross-comp.	0.64
2,000	Curtis horizontal	0.098	Vert. cross-comp.	0.36
5,000–8,000	Westinghouse-Parsons	0.100	Manhattan	0.48
5,000–8,000	Curtis horizontal	0.060	Vert. three-cyl. comp.	0.20
5,000–8,000	Curtis vertical	0.040		

A ratio of the average space occupied by the engine units to that of the turbine units reported is 4.7 to 1.

A similar table has been compiled from a paper by F. W. Dean[1] based on real output of the units at an 80 per cent. power factor. Besides the generating units, the condensing apparatus and exciters are included.

COMPARATIVE SPACE PER KILOVOLT-AMPERE

Capacities, kw.	Turbines		Engines		Ratio engines to turbines
	Dimensions, ft.	Sq. ft. per kv.a.	Dimensions, ft.	Sq. ft. per kv.a.	
500	14 × 6	0.168	18 × 26	0.935	5.57
1,000	14 × 8	0.112	24 × 30	0.720	6.43
1,500	19 × 9	0.114	26 × 35	0.605	5.30
2,000	21 × 9	0.094	28 × 37	0.518	5.51

Although the possibilities of great space reduction by installing turbines in place of reciprocating engines are clearly shown by the figures given above, it is peculiarly interesting to note that the present-day tendency toward less-crowded conditions in central stations makes the actual space reduction per kilowatt of plant rating less real than generally supposed as shown by the following tables compiled from published[2] data from 23 well-known central stations.

1. Capacity, ult. kw.
2. Boiler capacity, ult. hp.
3. Boiler hp. per rated kw.
4. Square feet per kw. (ground-floor plan).
5. Square feet per kw. (total single-deck plan).
6. Square feet per kw. (total generating room).
7. Square feet per kw. (net gen. room exc. switchboard).
8. Square feet per kw. (boiler room actual floor plan).
9. Square feet per kw. (boiler room total single-deck plan).
10. Square feet per boiler hp., boiler room (single-deck plan).

[1] Paper before National Association of Cotton Mfgrs., Boston, April, 1913.

[2] "Recent Developments on Steam Power Station Works," by J. R. BIBBINS, paper before A. S. and T. Ry. E. A., 1907 convention.

	18 Turbines			5 Corliss, vertical and horizontal engines			Ratio $\frac{F}{C}$
	A, Max.	B, Min.	C, Avg.	D, Max.	E, Min.	F, Avg.	
1	77,500	3,000	30,000	70,000	38,500	50,000	
2	62,500	2,500	22,500	43,000	23,000	33,500	
3	0.948	0.435	0.71	0.833	0.614	0.67	0.945
4	3.46	0.817	1.84	1.985	0.958	1.44	0.783
5	3.46	1.206	2.04	3.01	1.33	2.24	1.10
6	1.71	0.30	0.81	1.16	0.634	0.82	1.01
7	1.15	0.208	0.59	0.784	0.524	0.65	1.10
8	1.746	0.49	1.13	1.063	0.384	0.66	0.585
9	1.746	0.679	1.28	2.12	0.768	1.42	1.110
10	2.65	0.778	1.77	2.75	1.08	2.10	1.19

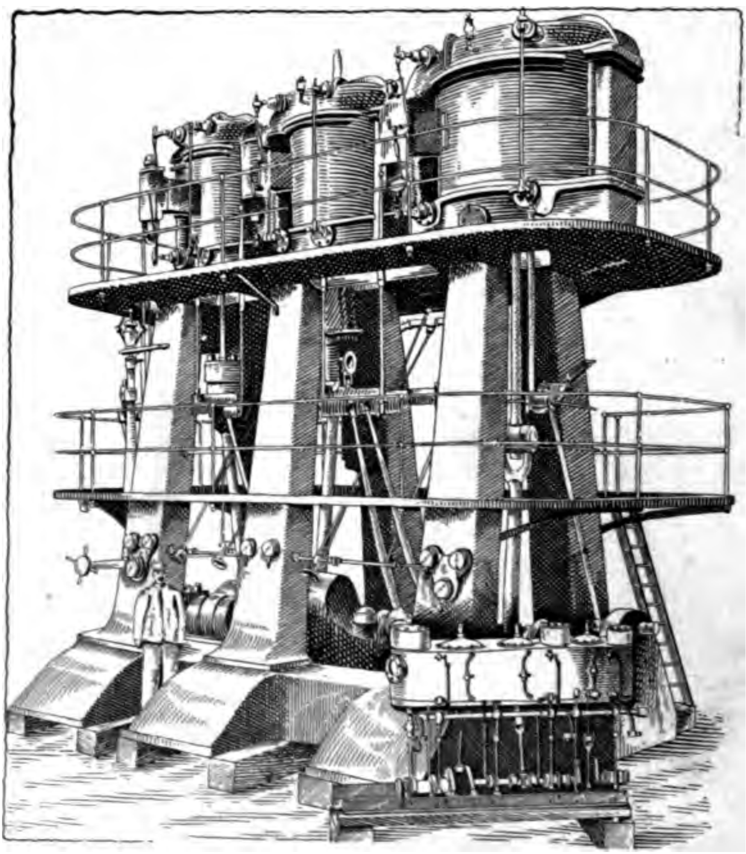


FIG. 38.—Comparative size of 1000-hp. pumping engine and 2500-hp. torpedo-boat engine.

Engine Flywheels.—Flywheels are necessary in the majority of installations. Marine engines require no flywheels, or rather the water-wheel and propeller serve this purpose. The locomotive requires no flywheels since the driving wheels and the living force of the engine and train serve this purpose. When there are two cranks at 90°, or three at 120°, the weight of the flywheel diminishes rapidly. For rough work in rolling mills, etc., with quartering cranks the flywheel is often dispensed with to facilitate quick reversal of the engine.

The size of the wheel depends upon the regulation required of a given engine. A variation of 5 per cent. in the speed is often allowable in factory engines while in certain types of electric lighting stations the allowable variation is only 1 per cent. On large, important 60-cycle installations a variation either side of perfect rotation of one-fourth of a geometrical degree has been specified.

A fair guarantee to ask is that the speed of the engine shall not vary more than from 2 to 2½ per cent. above or below the normal speed under any conditions of load.

First Cost vs. Economy of Operation.—The first cost of engines of the different types in a measure varies inversely as the economy and durability, but as a rule the saving in the coal bill due to the operation of engines of the better grade will more than pay the excess in the first cost during the first few years of operation, which in some respects might be considered as paying interest on the investment.

It is not always true that plants containing the most expensive engines have as a whole the highest initial cost, as the more economical type of engine requires less boiler capacity, and the saving in boiler-room cost may be enough to cover the extra engine cost.

For convenience it may be stated that a boiler horsepower requires the evaporation of approximately 30 lb. of water at the usual temperatures of feed water and at ordinary steam pressures.

As already seen the amount of steam required by different engines varies from about 10 lb. per horsepower-hour in best practice to 50 or 60 in poor grades of engines.

This would indicate that with the most economical types of engines 1 boiler horsepower would be sufficient to supply 3 engine horsepower, while with the poorer types 1 boiler horsepower would supply steam for about ½ an engine horsepower. In one case the boiler would have to be of six times the capacity of the other to supply the same amount of power.

This may be put in another form as follows: 1 boiler horsepower = 33,480 B.t.u. per hour.

In reasonably large plants an indicated horsepower = 12,000 to 24,000 B.t.u. per hour.

Then in general in large plants it is sufficient to provide only enough boiler horsepower to equal one-half the engine indicated horsepower as this will not only meet the normal engine demand for steam, but will give sufficient margin to cover the cutting out of boilers for cleaning and repairs.

For small amounts of power or for intermittent use inexpensive engines will prove satisfactory, but when the cost of power is a large item and when the engine is run continuously, the best is none too good.

Engines are usually designed to give the rated power when working with the best ratio of expansion. The most economical use of steam is usually found, therefore, when engines are working under normal full load, provided they are working under favorable conditions. Engines must be properly proportioned for the load they are to carry, if high economy is to be maintained. The best results are invariably found with engines operating under steady load. In electric-power stations the load generally varies within wide limits and a number of tests of such stations shows that the same grade of engine consumes about 50 per cent. more steam for the same work than for service where the load is uniform.

Compounding is advisable for large units if the load is reasonably uniform. With a fluctuating load the simple engine governs better and is about as economical.

PROBLEMS

1. Engine is 12 in. by 18 in. Piston rod 2 in.; m.e.p. for head end = 40 lb. and for crank end = 37 lb.; 250 r.p.m. Find (a) Horsepower of head end. (b) Horsepower of crank end. (c) Total indicated horsepower (i.hp.).

2. Engine is 6 in. by 9 in. Piston rod $1\frac{1}{4}$ in.; m.e.p. for head end = 33 lb. and crank end = 30.5 lb.; 300 r.p.m. Find same as in problem 1.

4. A locomotive running at the rate of 45 miles per hour has 72-in. driving wheels and cylinders 18 in. by 30 in. Piston rod $2\frac{1}{2}$ in.; m.e.p. 100 lb. Find the indicated horsepower of the locomotive.

5. In problems, 2, 3 and 4, find the horsepower constant for both head and crank end of each engine.

6. An engine is required to indicate 37 hp. with m.e.p. = 40 lb., stroke = 18 in., r.p.m. = 90. Required the diameter of the cylinder.

7. Test the "Handy Rule" on page 19 in problems 2, 3 and 6.

8. A small factory has a 450-i.hp. simple, high-speed, non-condensing engine and a 450-i.hp. compound, low-speed, condensing engine. If the steam used by the two engines is allowed to waste, what will be the difference in the steam consumption of the two engines for a month of 26 days, when operating at full load for 10 hr. each day?

9. What size compound, high-speed, non-condensing engine would give the same total steam consumption as the 450 i.hp. simple engine in problem 8?

What size compound high-speed condensing engine would give the same total steam consumption?

10. In a recent engine test the following readings were secured: (a) When running non-condensing; length of run, 10 hr.; reading of feed-water meter at beginning of test,

26,958.7 cu. ft. and at end, 27,324.5 cu. ft.; average temperature of feed water at meter, 195°F.; voltmeter, 230; ammeter, 130. (*b*) When running condensing; time, 10 hr.; feed-water meter, 34,652.0 and 34,911.6; feed water, 120°F.; load as in (*a*). Find per cent. gain in water consumption per e.hp.-hr. by running condensing.

11. Given the following data for a steam pumping engine:

Diameter cylinders, inches	32	60	90
Diameter piston rods, inches	3	6	9
(rods pass entirely through cylinders)			
Stroke, feet	5		
M.e.p., pounds	68	19	8.5
Water pumped per 24 hr., gallons	36,000,000		
Head on pumps, or lift, feet	160		
Steam used per hour (containing 2 per cent. moisture) pounds	12,500		
Guaranteed duty, foot-pounds, per 1,000 lb. dry saturated steam	140,000,000		
Bonus, $1,000 per million ft.-lb. above	140		
Forfeiture, $2,000 per million ft.-lb. below	140		
R.p.m.	21.8		

Determine:

1. The indicated horsepower of the engine.
2. The water horsepower of the engine and the mechanical efficiency.
3. The consumption of dry saturated steam per hour per indicated horsepower and per water horsepower.
4. The bonus or forfeiture, if any.

12. The following is the operating performance of a two and three quarter ($2\frac{3}{4}$) million gallon centrifugal pumping unit consisting of two pumps (in series), gear-driven by a condensing steam turbine.

Discharge head = 158 lb. per square inch.
Suction lift = 12 ft.
Rate of pumping = 1,890 g.p.m.
Steam per hour = 3,965 lb.
Steam pressure = 150 lb. per square inch gage and dry saturated.
Exhaust pressure = 0.5 lb. per square inch absolute.

Determine:

1. The water horsepower;
2. Pounds of steam per water horsepower-hour.
3. Duty per million B.t.u.

How does this duty compare with the average performance of reciprocating triple-expansion pumping engine? What considerations might justify the installation of the lower duty centrifugal unit?

What brake horsepower rating should be specified for the turbine in the above case?

13. For purposes of preliminary estimate, determine the approximate size of building (square feet of floor area) for a steam-turbine installation of 3,000-kw. capacity.

(*a*) Turbine room;
(*b*) Boiler room;
(*c*) Plant.

14. Determine the heat economy, thermal efficiency, and efficiency ratio of the 169-hp. Buckeye-mobile described on page 53.

CHAPTER III

ELECTRIC GENERATORS AND MOTORS

Efficiency and Cost.—The discussion of electric transmission of power and of the relative merits of the D.C. and A.C. systems will be presented later. The efficiency and cost of this type of equipment are, however, presented at this point.

MECHANICAL EFFICIENCY OF GENERATORS. PER CENT.

Kw.	Load = 1/4	2/4	3/4	4/4
50	72.0	81.0	84.0	88.0
100	74.5	83.5	86.0	91.0
250	76.0	85.5	89.0	93.0
500	77.5	87.0	90.5	94.5
750	78.0	87.5	91.0	95.0
1,000	78.0	88.0	91.5	95.5
1,500	78.5	88.5	92.0	96.0
2,000	79.0	89.0	92.5	96.5
5,000	79.0	89.0	92.5	96.5

The efficiency of generators at other than full load may be found approximately from the following table.

EFFICIENCY OF GENERATORS IN PER CENT. OF FULL-LOAD EFFICIENCY

Per cent. of rated load	Per cent. of full-load efficiency
100	100.0
90	99.4
80	98.4
70	96.8
60	94.7
50	92.0
40	88.0
30	84.0

With direct-connected engine-generator sets in which the sizes of the two machines are properly proportioned to each other, the individual efficiencies may be combined, giving the following combined mechanical and electrical or the overall full-load efficiencies and relative efficiencies with fractional loads as indicated on p. 77.

A convenient method for the conversion of kilowatt generator rating to indicated horsepower engine rating is by combining into one factor the ratio $\frac{1,000}{746}$ (746 watts = one hp.), or 1.34, and the above overall efficiencies.

For example, the 50-kw. generator set will require at 75.3 per cent.

Full load		Fractional load	
Rating of generator, kw.	Per cent. overall efficiency of unit	Per cent. of the rated load on generator	Per cent. of the full-load overall efficiency of unit
50	75.3	100	100.0
100	78.5	90	98.5
250	82.8	80	96.1
500	85.4	70	92.6
1,000	87.7	60	89.0
1,500	89.5	50	84.5
2,000	91.2	40	79.4
3,000	91.5	30	72.5
		20	62.9
		10	50.8

overall efficiency, an engine indicated horsepower rating of $\frac{50 \times 1.34}{0.753} = 50 \times 1.78 = 89$ i.hp., 1.78 being the resultant conversion factor. These factors will be as follows for the several sizes listed.

CONVERSION FACTOR (FULL RATED LOAD)

Rating of generator, kw.	Factor
50	1.78
100	1.71
250	1.62
500	1.57
1,000	1.53
1,500	1.50
2,000	1.47
3,000	1.465

At fractional loads the factors must be increased with the decrease of the overall efficiency. The conversion factor corresponding to the unit rating may be divided by the proper per cent. of full-load overall efficiency or may be multiplied by its reciprocal. The reciprocals of the per cent. values in the above table, expressed as multipliers, are given below.

FRACTIONAL LOAD MULTIPLYING FACTOR

Per cent. load on generator	Factor
100	1.00
90	1.015
80	1.04
70	1.08
60	1.12
50	1.18
40	1.26
30	1.38
20	1.59
10	1.97

The approximate cost of generators and motors is contained in the following tables and diagrams:

DIRECT-CURRENT BELTED GENERATORS,[1] COST PER KILOWATT

Kw.	Slow-speed	Moderate-speed	High-speed
5	....	\$37.00 (1,200 r.p.m.)	\$30.00 (1,800 r.p.m.)
10	....	28.50 (1,000 r.p.m.)	23.00 (1,800 r.p.m.)
25	\$17.00 (900 r.p.m.)	16.00 (1,300 r.p.m.)	
100	12.30 (675 r.p.m.)	8.50 (100 r.p.m.)	

ALTERNATING-CURRENT BELTED GENERATORS

Kv.a.	Speed	Cost per kv.a.
15	1,800	\$22.50
25	1,800	18.00
100	900	14.00
500	360	10.00

ALTERNATING-CURRENT DIRECT-CONNECTED GENERATORS

Kv.a.	Speed	Cost per kv.a.
50	300	\$20.00
75	277	17.00
250	200	11.00
500	120	11.00
1,000	100	10.00

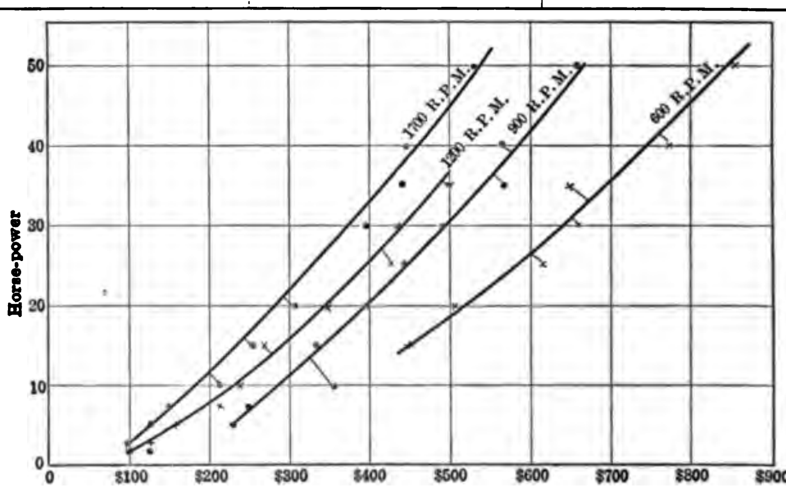


FIG. 39.—Cost of direct-current shunt and compound motors. 115 and 230 volts. (Prepared by L. B. Beatty, 1915.)

[1] These costs of generators per kilowatt are taken from the section on costs, "Machine Shop Electrical Handbook," by C. E. CLEWELL, McGraw-Hill Book Co., N. Y.

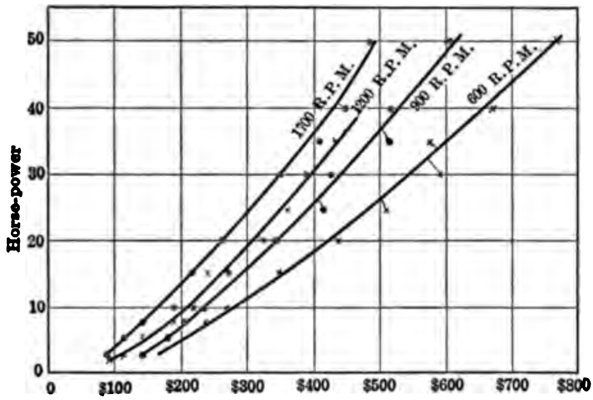


FIG. 40.—Cost of direct-current shunt and compound motors. 550 volts.
(Prepared by L. B. Beatty, 1915.)

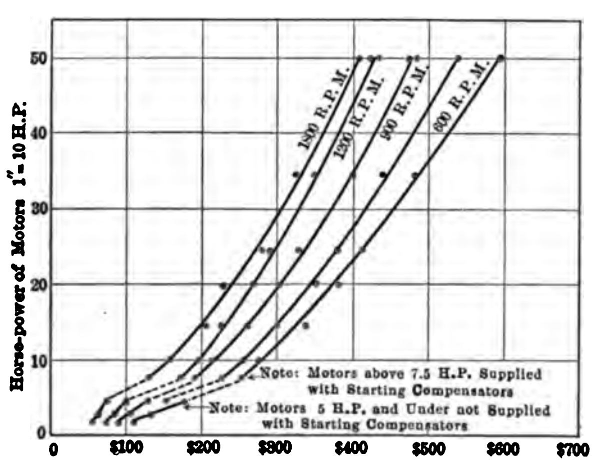


FIG. 41.—Cost of induction motors—squirrel-cage. 110 to 550 volts, 60-cycle.
(Prepared by L. B. Beatty, 1914.)

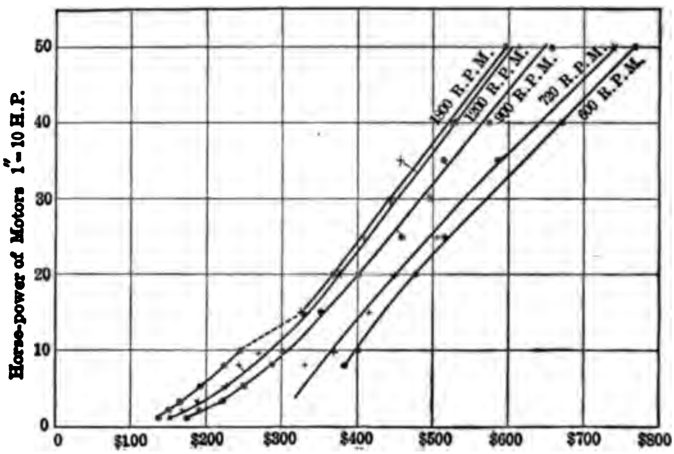


FIG. 42.—Cost of induction motors—phase-wound 110 to 550 volts, 60-cycle, 2-ϕ and 3-ϕ

(Prepared by L. B. Beatty, 1915.)

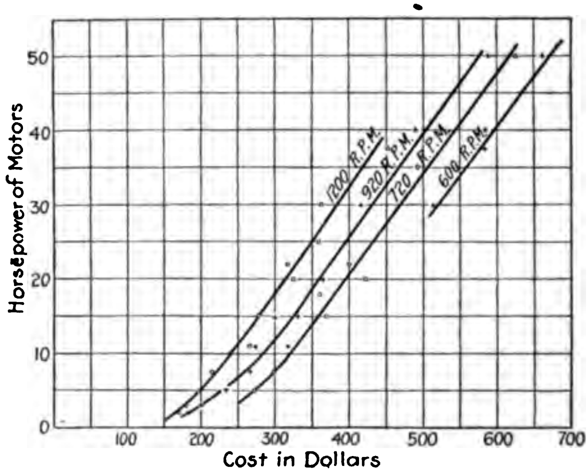


FIG. 43.—Cost of induction motors—phase wound. 110 to 550 volts, 60-cycle, 2-ϕ and 3-ϕ.

(Prepared by L. B. Beatty, 1915.)

PROBLEMS

15. The engine of problem 10 is a simple, high-speed unit.

(*a*) What was the approximate steam consumption per brake horsepower-hour under the two conditions of the test shown?

(*b*) What was the approximate steam consumption per indicated horsepower-hour under the two conditions?

16. A test of a compound Corliss engine with D.C. generator, direct-connected, gave the following readings when running non-condensing:

Length of run, 5 hr.

Water-meter readings, 72,850 and 73,500 cu. ft.

Voltmeter, 240; ammeter, 1,500.

What was the approximate steam consumption per indicated horsepower-hour?

17. The owner of a small factory has installed two simple high-speed non-condensing steam engines with direct-connected D.C. generators.

Engine *A* is guaranteed not to consume more than 63 lb. of dry saturated steam per kilowatt-hour at full load. At the time of the test the following data were obtained.

Cylinder diameter	= 10 in.
Stroke	= 15 in.
Diameter piston rod	= 2 in.
M.e.p. head end	= 40 lb.
M.e.p. crank end	= 42 lb.
R.p.m.	= 250
Total steam used per hour	= 2,100 lb.

Engine *B* is guaranteed not to consume more than 29 lb. of dry saturated steam per indicated horsepower-hour at full load. At the time of the test the following data were obtained.

Voltmeter reading	= 230
Ammeter reading	= 230
Total steam used per hour	= 2,850 lb.

1. Was engine *A* safely within the guarantee?
2. Was engine *B* safely within the guarantee?
3. Which engine showed the better economy? Approximately how much better?

18. The following are the full-load performances of a 500-kw. direct-connected engine and a 500-kw. turbo-generator operating under conditions as noted:

	Steam consumption, lb. per hr.	Initial pressure, lb. per sq. in. gage	Vacuum, in. of Hg.	Superheat °F
Engine ...	13,750	125	25.5	(dry)
Turbine...	10,700	150	28.5 (30-in. barometer)	100

Determine for each and compare:

(*a*) The steam economy in pounds per kilowatt-hour.

(*b*) The heat economy in B.t.u. per kilowatt-hour.

(*c*) The thermal efficiency in per cent.

(*d*) The heat economy of a Rankine cycle for the conditions given, in B.t.u. per kilowatt-hour.

(*e*) The efficiency of a Rankine cycle for the conditions given, in per cent.

(*f*) The efficiency ratio.

CHAPTER IV

FOUNDATIONS

1. Must support concentrated weight of engine upon the ground by distributing the weight over sufficient area to prevent settling.

2. Must go far enough below surface to be beyond settling either from frost, vibrations, or influence of loads borne by adjacent ground. Rarely safe to permit a depth less than 3 ft. below the general level. In cold regions effect of frost is felt down to a depth of 6 ft. The foundations for small engines are rarely less than 4½ ft., while for large units the depth is sometimes as great as 20 to 25 ft. Engine foundation depths are usually decided by other considerations than weight, such as location of auxiliaries, cellar or basement, etc.

3. Must have mass and weight enough to hold engine still against unbalanced forces.

4. Must have mass enough to take up vibrations if bedplate is not massive enough.

The rules for allowable weight per square foot on different soils vary in different cities, but in general the supporting power in tons per square foot may be taken as (Baker, "Treatise on Masonry Construction"):

Rock—granite, etc., in hard compact strata	200 to —
Rock—limestone, equal to best masonry	25 to 30
Rock—sandstone, equal to best brick masonry	15 to 20
Rock—broken and well compacted	7 to 20
Rock—soft and pliable as shale, equal to poor brick masonry	15 to 20
Hard pan—gravel and sand, well cemented with clay	8 to 10
Clay—thick beds and dry	4 to 6
Clay—thick beds and moderately dry	2 to 4
Clay—soft, wet, confined	1 to 2
Gravel—coarse and dry, well compacted and confined	8 to 10
Sand—dry, compact, well cemented with clay	4 to 6
Sand—clear and dry, confined in natural beds	2 to 4
Quicksand—marshy and alluvial soils, etc., confined	0.5 to 1

If the soil is of low bearing power, piling must be used. Formerly piles were usually of spruce or hemlock. At the present time yellow or red pine, oak, birch and beech are used. Steel piles and concrete piles are also meeting with favor.

Wooden piles are at least 5 in. in diameter at the point and 10 in. at the butt for piles 20 ft. or less in length; 6 in. at the point and 12 in. at the butt for piles over 20 ft. long.

The "*Engineering News* Pile Formula," often used, is, safe bearing power in tons = twice the weight of hammer in tons multiplied by height of fall in feet divided by one + penetration of pile in inches under last blow.

Often the tops are cut level and the soil excavated for a foot. Concrete is then filled in over the heads of the piles, sometimes to a depth of several feet.

The Metropolitan Street Ry. 96th Street power house, New York City, and the Kent Avenue Power House, Brooklyn, are on 8-ft. beds of concrete over piles 30-in. centers.

Engine Foundations Proper.—The engine builder always furnishes an engine foundation plan, showing the various footings which must be supported and the location and sizes of the various anchor bolts. It is customary to build a wooden template covering the entire foundation and supporting square wooden boxes about 1½ in. larger internally than the diameter of the foundation bolts. These act as forms and when removed from the foundation allow plenty of play for the foundation bolts. The heavy cast-iron washers or anchor plates are set in the concrete form at the base of these boxes and pockets are provided below them so that the nut on the lower end of the foundation bolt may be reached by a wrench. At the present time foundations are always built of concrete in monolithic masses as far as possible, and the foundation is usually allowed to set a week or 10 days before any heavy weights are placed upon it.

Good concrete is made by mixing 1 part of good Portland cement, 3 parts of sand and 5 parts of broken stone or gravel, the latter small enough to pass through a 2-in. ring. This should be mixed very wet.

Another proportion sometimes used for engine foundations is 1 :2 :4.

Concrete foundations weigh approximately 150 lb. per cubic foot.

Cost of Concrete Foundations.—Large foundations or foundations of the simplest form may be put in for from $6 to $8 per cubic yard.

If the foundations are small or irregular, requiring special forms and considerable template work the price will often be about double the above figures or $13 or $14 per cubic yard complete, including all excavating and carpenter work.

Another basis of estimating foundation costs is:

Excavation without shoring in soft material	50 cts. to $1 per cu. yd.
Excavation without shoring in rock	$1 to $4 per cu. yd.
Concrete	$6 to $7 per cu. yd.
Forms	15 cts. per sq. ft.
Waterproofing (if used)	40 cts. per sq. ft.

Average cost figures for concrete work of a large project recently reported[1] are:

Group 1.—For plain concrete used for walls, approaches, bins, conduits, etc.

Group 2.—Miscellaneous concrete foundations.

[1] See "Unit Construction Costs" by E. H. Jones, McGraw-Hill Book Co.

Group 3.—Reinforced-concrete walls, foundations, sumps, etc.
Group 4.—Items 1, 2 and 3 combined.

Group No.	Total amount, cu. yd.	Labor cost per cu. yd.			Material cost per cu. yd.			Total cost per cu. yd.		
		Max.	Min.	Avg.	Max.	Min.	Avg.	Max.	Min.	Avg.
1	7,779	$8.07	$0.75	$2.85	$8.82	$3.37	$4.82	$16.11	$5.53	$7.67
2	8,706	6.85	2.01	2.99	8.90	3.42	5.36	13.52	6.00	8.36
3	2,830	7.49	3.40	3.37	9.48	6.42	5.48	16.35	10.35	13.53
4	19,315	8.07	0.75	3.37	9.48	3.37	5.48	16.35	5.53	8.85

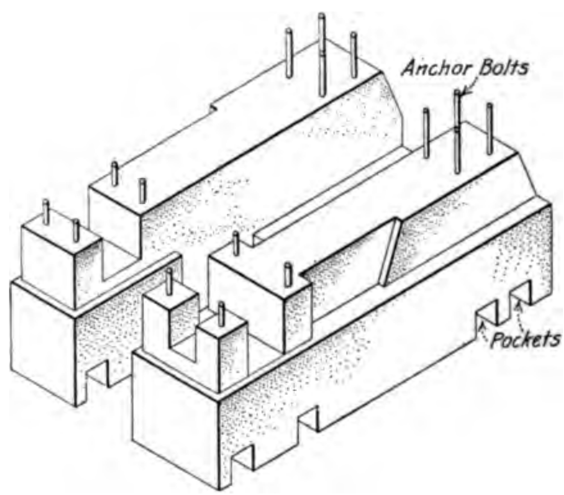


Fig. 44.—Foundation for cross-compound engine

A consulting engineer of large experience shows the cost of all the foundations required in steam-power plants to be approximately as follows:

Cost of Foundations per Engine Horsepower

For Simple Non-condensing:									
I.hp. of plant	10	12	14	15	20	30	40	50	75
Cost per horsepower	$5.70	$5.50	$5.40	$5.35	$5.25	$5.15	$5.05	$4.90	$4.60
For Simple Condensing:									
I.hp. of plant	10	12	14	15	20	30			
Cost per horsepower	$8.50	$8.40	$8.30	$8.10	$7.80	$7.40			
I.hp. of plant	40	50	75	100					
Cost per horsepower	$7.00	$6.70	$6.00	$5.70					
For Compound Condensing:									
I. hp. of plant	100	200	300	400	500	600			
Cost per horsepower	$5.70	$5.60	$5.50	$5.40	$5.30	$5.20			
I.hp. of plant	700	800	900	1,000	1,500	2,000			
Cost per horsepower	$5.10	$5.00	$4.90	$4.80	$4.40	$4.10			

Foundation Bolts.—Foundation bolts for small engines are rarely below $\frac{3}{4}$ in. in diameter and from this they increase in size with the engine and the stresses to $2\frac{1}{2}$ to 3 in. on large vertical engines up to 10,000 hp. These bolts may be of medium steel and the larger sizes always have upset ends with the threads of a larger size than the body of the bolt. They should not be too long on account of temperature and stress changes, but should run down far enough into the foundations to get sufficient weight of concrete between the engine bed and the anchor plate. The anchor plates are always of cast iron, designed after the manner of column bases and usually have lugs cast on their bottom surface to hold the lower nut.

It is not customary to use locknuts on anchor bolts, but they are sometimes used, especially for vertical engines on those bolts which pass through the bearings and are also used to hold down the bearing caps.

Grouting.—The bedplate, after being placed on the foundation, is lined up and leveled by means of shims and steel wedges. After this is done the anchor bolts are inserted and the nuts tightened up hand-tight, using a proper wrench for the size of the nut. This leaves a joint between the bedplate and the foundation, which varies from $\frac{1}{2}$ in. in small engines to 1 to $1\frac{1}{2}$ in. in large engines. A clay dam is now built around the bedplate and cement grout, usually neat or at worst 1 part cement to 2 parts sand, is poured into this space through holes which are left for this purpose in the bedplate. Where the bedplate is hollow it is quite customary to fill it with this cement grout. Care must be taken that there are no air bubbles left under the bedplate and that the cement runs out until held by the dam on all sides of the bedplate. The grout is now allowed to set for 3 or 4 days, the dam and the ragged edges of the grout chipped away and then the foundation bolt nuts are tightened to full bearing by sledging, taking care that the alignment and the level of the bedplate are not changed.

Other materials have been used instead of the cement grout, such as a rust joint made out of iron filings and sal ammoniac, melted sulphur, type metal, oakum, felting and in some cases even wooden wedges, but at the present time practically the only material used is the cement grout.

Alignment.—If the bedplate is in one casting, as is usual in small engines, it is only necessary to level up the planed surfaces of the guides in two directions at right angles to each other, and even this is not absolutely necessary. Where the bedplate is in two or more parts the problem becomes a much more difficult one. The various sections of the bedplate are assembled on the foundations in approximately their final positions. They are then bolted together, or the T-headed links are heated and placed in the holes provided for them, thus bringing the parts of the bedplates into close contact by the shrinking of the links. The bedplate, then, as a whole is wedged up until it is level in both directions.

It is frequently necessary during this period, especially in vertical engines, to have one part of the bedplate slightly higher than its final position in order that the deflection caused by the added weights placed on the plate may just bring the bedplate to a true level. This work will be greatly expedited by the use of an engineer's level, although on smaller engines it is customary to level and center the bedplate from wires which have been stretched through the final axis of the cylinder and shaft. It should be remembered that with large shafts and with considerable distance between the bearings there will be measurable deflection, in which case, with vertical engines, the cylinders will not be set in the vertical plane, but will be inclined so that the cylinder axis will be at right angles to the shaft in its deflected position. The cylinders of a 5,000-hp. cross-compound vertical engine, with the flywheel and generator between the cylinders may be as much as $\frac{5}{16}$ in. closer together than the centers of the cranks: In one case on an engine of this size where the generator and flywheel were outboard of the engine, the outboard bearing had to be set nearly $\frac{5}{8}$ in. above a true level to produce quiet and cool running.

PROBLEMS

19. Estimate the size and cost of foundations for the following steam engines.

(*a*) 50-hp. simple high-speed.

(*b*) 500-hp. tandem compound Corliss.

(*c*) 2,000-hp. Manhattan-type Corliss compound.

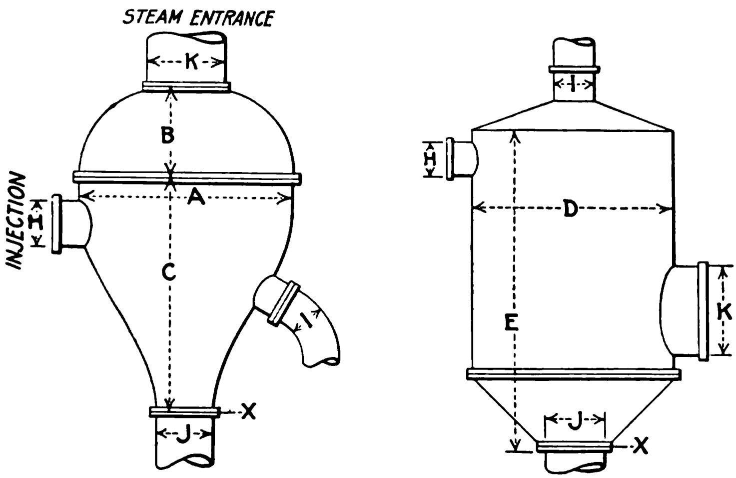
20. The dimensions of bedplate of a 1,000-kw. turbo-generator are 6 ft. 6 in. by 17 ft. 0 in. The distance from basement floor to turbine-room floor is 14 ft. The soil below basement level is moderately dry clay. Sketch a proposed foundation for the turbine and estimate the cost of the same.

21. Eight steam-engine and generator units of 2,500 kw. each, weighing with auxiliaries 350 lb. per indicated horsepower, are to be erected on a pile foundation. The area covered is 100 ft. by 100 ft. Piles, 4 ft. center to center. In driving the piles a 2,000-lb. hammer was used; drop, 10 ft.; last penetration of pile, 1 in. Is the foundation safe? If so, how much leeway is there for each pile, in pounds? If not safe, how much excess load is there for each pile, in pounds?

CHAPTER V

CONDENSERS

Condensation may be considered as of two kinds; mixed condensation when the steam and cooling water are brought together in the same vessel or machine as in the jet, barometric and ejector types, and surface condensation when a film of metal prevents mixing as in the surface and atmospheric types.



Parallel-flow type — Counter-current type

FIG. 45.—Barometric condenser heads.

Mixed Condensation.—

Let t_s be the temperature of the steam to be condensed.
t_o be the temperature of the injection water.
t_1 be the temperature of the outlet or hotwell water.
H be the total heat in the steam at t_s.
h be the heat in the liquid at t_1.
w be the pounds of steam per hour to be condensed.
Q be the pounds of water per hour needed for condensing.

Then $\dfrac{Q}{w} = R = \dfrac{H - h}{t_1 - t_o}$ (1) and R is the ratio of water to steam required for condensation.

t_1 theoretically is equal to t_s but in practice t_1 is from 5° to 10° lower than t_s owing to the presence of air and imperfect mixing. In proportioning ordinary jet or barometric condensers w is the normal amount of steam to be condensed and a 50 per cent. overload is common at some reduction of vacuum.

Let G = the cubical contents of the cone in cubic feet.

Then $G = 0.00143w + 8.25$ cu. ft. (2)

The allowable velocity in the tail pipe is 5 ft. per second.

Then $J = 0.073\sqrt{w}$ in. (3)

$A = 15.7\sqrt{G}$ in. (4)

$B = 0.3A$

$C = 1.2A$

$$H = \begin{cases} J-1 \text{ for small sizes.} \\ J-2 \text{ for large sizes.} \end{cases}$$

$I = \frac{J}{3}$ for 26–27 in. Increase slightly for 28 in.

Steam velocity in K about 600 ft. per second.

$$K = \sqrt{\frac{w}{50}} \text{ abt.} \tag{5}$$

The height of the flange x above the level of the hotwell should never be less than 35 ft. and may be greater to advantage.

Size of condenser, lb. of steam per hr.	5,000	10,000	15,000	20,000	25,000	30,000	40,000	50,000	60,000	80,000	100,000
Exhaust K	10"	14"	17"	20"	22"	24"	28"	32"	35"	40"	45"
Tail pipe J	6	8	9	10	11	12	14	16	18	20	24
Injection H	5	7	8	9	10	11	12	14	16	18	20
Air I	2	3	3	3	4	5	5	6	6	7	8
Diameter A	38	45	48	52	56	60	63	68	72	78	82
B	9	13	14	16	17	18	19	20	21	23	25
C	45	54	57	62	67	72	75	81	86	94	99
D	28	33	36	38	40	42	46	50	52	57	60
E	56	66	72	76	80	84	92	100	104	114	120

The condenser bell should be as near to the exhaust flange as possible as friction and velocity head count up very fast with good vacua. The barometric pipe may be replaced by a pump of some kind. In the ordinary jet condenser the bell is placed over the suction chamber of the pump. In all arrangements of this type there must always be a sufficient head of water over the suction valves to ensure their rising.

Ejector condensers follow the same principle as the ordinary type when a tail pipe is used and the throat of the ejector is usually figured for a velocity of 15 to 20 ft. per second.

Here $K = \sqrt{\frac{w}{50}}$ as before, inches

$H = 0.073\sqrt{w} - 1$ in.

$T = 0.6H$ in.
$J = 0.073\sqrt{w}$ in.
$O = 5K$ about

The flange X should be about 40 ft. above the level of the tail water.

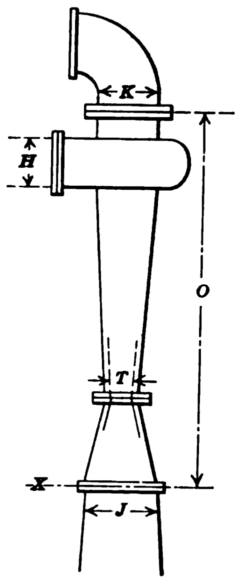


FIG. 46.—Ejector condenser, Bulkley type.

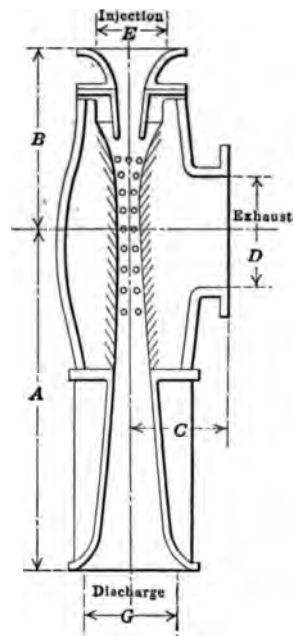


FIG. 47.—Ejector condenser, Schüte & Koerting type.

SCHUTTE EDUCTOR CONDENSER

Pounds steam per hour	A	B	C	D	E	G
520	9	6¼	3¼	2	1¼	1¼
1,040	12½	8	4⅛	3	2	2
2,240	18½	11	5	4	3	3
3,300	21½	13	5½	5	3½	3½
4,800	25¾	15	6½	6	4	4
6,600	30½	17	7¼	7	4½	4½
9,000	35½	19	8	8	5	5
12,000	41	21½	9	9	6	6
15,000	46½	23½	10	10	7	7
21,000	53	27	11	12	8	8
28,500	61¾	30	12	14	9	9
36,000	70	34	14	16	10	10
24,000	80	38	15	18	12	12
60,000	90	43	18	20	14	14
90,000	108	51	21	24	16	16

Ejector condensers without the tail pipe are common and when properly installed and operated work very well.

Condenser bells of most any shape may be used and are equally efficient if the water and steam are brought into contact and the air is collected and carried away. This may be done by a separate dry air pump, or the air pipe may be led into the throat of the tail pipe.

Surface Condensation.—The steam is condensed on the outer surface of metallic tubes through which the condensing water flows.

Let N = total heat to be transmitted per hour, B.t.u.
S = outside surface of tubes, total in square feet.
θ_m = mean temperature difference of water and steam, °F.
K = coefficient of heat transmission, B.t.u. per square foot per hour, per °F., diff. in temperature.

Then $N = K\theta_m S$ $$Q = w\frac{H - h}{t_1 - t_o} \qquad (6)$$

and $N = w(H - h)$ $$S = \frac{Q(t_1 - t_o)}{K\theta_m} = \frac{w(H - h)}{K\theta_m} \qquad (7)$$

Let t_s = vacuum temperature.

For practical work K may be taken as constant for any one condition, although it has been shown by experiment to be subject to small variations with θ_m.

The mean temperature difference for rough calculation with small rise in temperature of the circulating water may be the arithmetical mean without serious error, but for most calculation the geometrical mean should be used.

$$\theta_m = \frac{t_1 - t_o}{\log_\epsilon \dfrac{t_s - t_o}{t_s - t_1}} \qquad (8)$$

The quantity of circulating water Q is a function of the number and size of the tubes, the number of water passes in the condenser and the velocity of the water in the tubes. The values of K depend on the velocity of the water also as well as on the material of the tube, its cleanliness and the richness of the steam and air mixture in the condenser.

The general formula for K is $K = kcp^2U\sqrt{V_w}$ (9)

Where k equals 350 a constant,

c equals the cleanliness coefficient varying from 1.0 to 0.50,

p equals the air richness ratio $\frac{P_s}{P_t}$.

U equals the material coefficient.

1.0	for copper.
0.98	for admiralty mixture.
0.97	for admiralty mixture oxidized.

0.95 for Muntz metal.
0.92 for aluminum bronze.

V_w = water velocity in tubes, feet per second.

The water velocity should be about 8 ft. per second. The material coefficient may usually be taken at 0.95 and the cleanliness coefficient at about 0.9 for such waters as New York or Chicago. The air richness coefficient is exceedingly difficult to measure experimentally but for tight condensers with efficient air pumps it may be taken at from 0.95 to 0.97. Under these conditions $K = 782$. The value of $K = 782$ is thus under the best conditions and should not be taken for design since tight condensers and air pumps are not the rule but the exception and tubes soon oxidize or become coated with dirt and scale. In commercial work $K = 350$ seems to be the usual figure but values as high as 600 have been guaranteed. It should be remembered that a surface condenser is rarely tested to its limit.

In condenser design the given quantities usually are w, t_o and the required vacuum. It is important that the place of measurement of the vacuum should be stated and this is usually in the nozzle connecting the prime mover to the condenser. The best vacuum will always be found at the air-pump suction, less in the body of the condenser, and the worst in the nozzle. The vacuum inside the prime mover will be less by the velocity head necessary to give motion to the exhaust and by friction in the nozzle. The allowable velocity in a turbine nozzle is about 600 ft. per second.

Starting with the vacuum in the nozzle certain assumptions must be made; first, the loss in the condenser known as drop—this in a well-designed condenser should not exceed 0.2 in. of mercury and t_s should be taken as the temperature corresponding to this reduced vacuum. For good practice t_1 should be from 8° to 10°F. lower than t_s and $t_1 - t_o$ is now known. H and h are known from the steam tables and Q may be calculated. $\frac{Q}{w}$ the ratio of cooling water to condensed steam usually ranges from 50 to 100 and it should be remembered that a large ratio means more power required in the circulating pumps.

Having t_s, t_o and t_1 the mean temperature difference may be calculated from (8) and the surface from (7).

Small tubes are best for the transmission of heat but cannot be used with dirty water so that the usual sizes are ⅝ in., ¾ in., 1 in. and in some cases with very bad water 1¼ in. or even larger.

Let a = area of tube in square inches.
l = length of tube in feet (sum of all passes).
d = diameter of tube in inches.

n = number of tubes in one pass.
A = area of one pass in square feet.
f = number of passes.

Then

$$n = \frac{144A}{a} = \frac{Q}{1{,}560aV_w} \tag{10}$$

and

$$l = \frac{3.82S}{nd} \tag{11}$$

The length of a single tube is $\frac{l}{f}$ and the tube ratio is $\frac{l}{d}$. This should be between 30 and 50. The best value of this ratio has not been established by experiment.

Some adjustments may be necessary due to space and other considerations which can be made at this time.

Tube spacing is important as there must be room for the glands and sufficient metal between them for strength. The minimum allowable spacing or pitch of tubes is

5/8-in. tubes	15/16-in. pitch	192 tubes per square foot.
3/4-in. tubes	1 1/16-in. pitch	147 tubes per square foot.
7/8-in. tubes	1 1/4 -in. pitch	106 tubes per square foot.
1 -in. tubes	1 3/8 -in. pitch	88 tubes per square foot.
1 1/4-in. tubes	1 5/8 -in. pitch	63 tubes per square foot.

The number of tubes per square foot of tube-sheet surface is given roughly by $n = \frac{166}{(\text{pitch})^2}$.

Glands should be of the same metal as the tubes and be provided with an inside lip to prevent creeping of tubes. The entrance of the gland should be rounded.

Rubber rings are much used for packings on European condensers with fresh condensing water, but the screw gland with corset-lace packing put in with an automatic gun is probably best. Tube packings may be fiber, woven hose or corset lacing. No animal or vegetable fats should be used on the packing as they form soluble compounds with copper, paraffine is the best wax to use with woven packings.

Tube sheets should be Muntz metal or brass. A rolled sheet will give the best service although cast sheets are used. The thickness of tube sheets should be 1/8 in. to 3/8 in. larger than the tube diameter.

Condenser shells are usually of cast iron ribbed outside against collapsing pressure but may be of steel plate or sheet brass (navy practice) stiffened with angles. Tubes should be supported at distances of 60 to

70 diameters by supporting plates usually of cast iron, drilled with $\frac{1}{16}$-in. clearance around the tube.

Water boxes should be large and designed to offer as little friction as possible to the passage of the water. A hole $\frac{1}{8}$ in. in diameter in the partition will allow the upper box to drain when not in use and the condenser is usually set on a slope of 1 in. in 15 ft. so that the tubes may drain. Where possible the steam should enter from the top and water at the bottom (counter-current principle) but this is not essential as parallel flow condensers give good results.

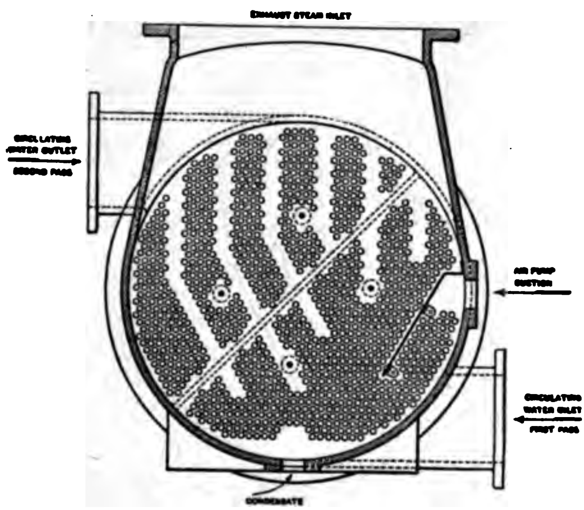


Fig. 48.—Cross-section spiroflow surface condenser.

The bottom of the circulating water outlet should be above the highest point of the tube bank. If this cannot be done at the water box the pipe should be carried up to the same height away from the condenser.

The steam passage should be direct to the tube bank and if possible the nozzle should be spread so no dead spots may be left away from the path of the steam. The upper bank of tubes may have a wider spacing than the lower or channels may be left open into the tube surface to afford a free passage for the steam. Baffle-plates and guide plates are also used for the same purpose but are not as efficient.

The steam flow should be directed to the coldest part of the condenser and here the dry air suction should be taken out. The suction should be screened to prevent water being carried into it.

Water connections should be figured for a velocity of 10 ft. per second and the air connection should be at least twice the hotwell water size which should be figured for about 6 ft. per second.

Owing to the wide range of steam consumptions for engines no definite relation exists between engine horsepower and required condenser surface. Similarly no definite relation exists between the condensing surface and the amount of steam condensed unless the cooling water temperature is constant. An average figure commonly quoted is 10 lb. of steam condensed per square foot of condensing surface for 24 to 26-in. vacuum with 70°F. cooling water.

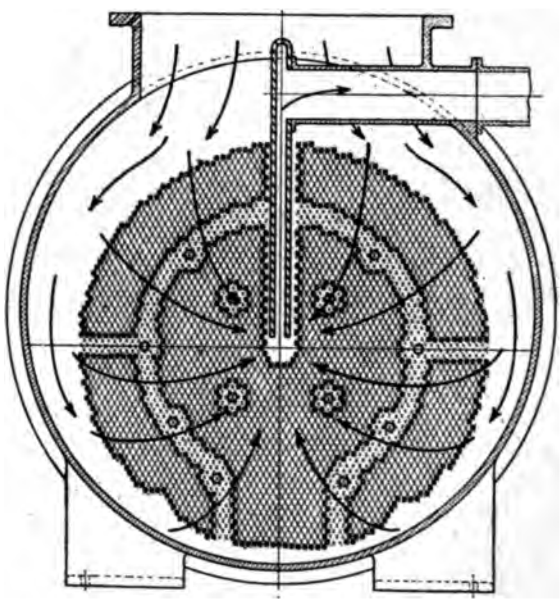


FIG. 49.—Cross-section Westinghouse surface condenser.

Although the circulating water required per pound of steam condensed varies widely in practice, depending on the vacuum maintained, the difference between the temperature of the steam due to the vacuum and the temperature of the condensate leaving the condenser, and the amount of air in the condenser, yet the following figures will serve as an indication of the variation in the amount of circulating water required due to differences in initial temperature of the circulating water.

Pounds circulating water per pound steam condensed (= R)

$$R = \frac{H - h}{t_1 - t_o} \qquad t_1 = t_e - 5$$

Vac.	$H-h$	t_e	t_1	R			
				$t_o = 50$	$t_o = 60$	$t_o = 70$	$t_o = 80$
24.0	1012	141	136	11.8	13.3	15.3	18.0
25.0	1017	133	128	12.0	15.0	17.5	20.8
26.0	1022	125	120	14.6	17.0	20.5	25.6
27.0	1028	114	109	17.5	21.0	26.4	35.4
28.0	1036	100	95	23.0	29.6	41.5	69.0
28.5	1041	90	85	29.8	41.7	69.5	209.0
29.0	1049	77	72	47.5	87.0	525.0	
	$t_1 = t_e$						
24.0	1012	141	141	11.1	12.5	14.3	16.6
25.0	1017	133	133	12.3	13.9	16.2	19.3
26.0	1022	125	125	12.6	15.7	18.6	21.7
27.0	1028	114	114	16.1	19.0	23.4	30.2
28.0	1036	100	100	20.7	25.9	34.5	52.0
28.5	1041	90	90	26.0	34.7	52.0	104.0
29.0	1049	77	77	38.8	61.5	150.0	

Surface condensers should be used when suitable boiler feed water is not easily obtained. If, however, suitable water for the boilers is easily obtained, the jet condenser should be used. It is simple and relatively inexpensive.

Condensing Apparatus for Turbines.—It has been definitely proved that the reciprocating engine is ill adapted to utilize a vacuum higher than 26 in. The turbine, on the contrary, can easily utilize vacuums up to 28 in. and with proper design vacuums of 29 in. and even 29½ in. may be taken care of to advantage. The condensing apparatus necessary to furnish a 28-in. vacuum will be nearly double the size of that required for 26-in., and for 29-in. vacuum will be larger still. The ratio of condensing water to condensate, instead of being 25 at 26 in. vacuum, varies from 80 to 110 at vacuums above 29 in., thus largely increasing the size of the circulating pumps and water piping. The hotwell pump is small in any case and can be neglected, but where reciprocating air pumps are used the size for 29 in. will be largely in excess of that for 28 in., while with the engine condenser no dry air pump is required. If rotary air pumps of the Le Blanc type are used, the space occupied will be about half the size of the circulating pump.

With the larger turbines of the present day and basements 20 to 25 ft. in the clear, it is possible to arrange all of this condensing apparatus under the space occupied by the turbine unit and the necessary aisles around it. This will not be possible if a concrete foundation is used, and in most of the modern installations the turbine is supported on a structural-steel foundation which at the same time supports the condensers, condenser piping and operating room floor (see *Engineering News*, January 14, 1915, p. 66).

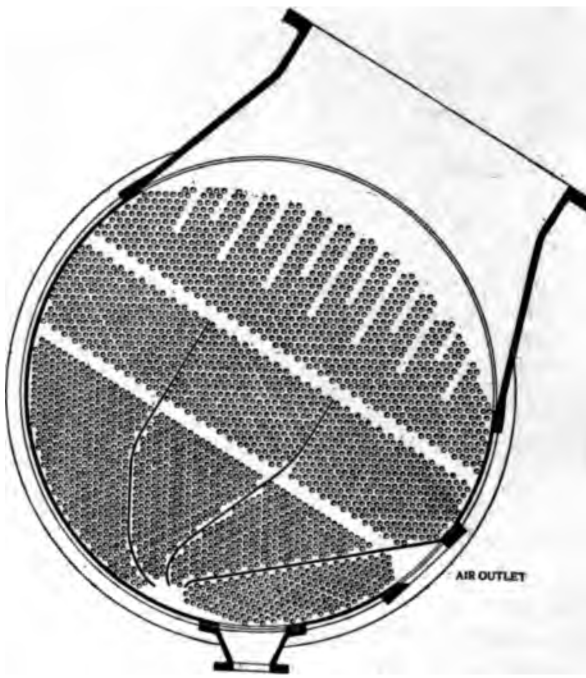


FIG. 50.—Cross-section contraflo surface condenser.

Air Condensers.—Condensers in which air is used instead of cooling water are not common in steam engineering but are used where water is scarce and bad. The steam to be condensed is led into pipes or chambers of thin plate around which the air is circulated either by a chimney or fan. A notable condenser of this type was installed at Kalgoorlie, Australia, to condense 32,000 lb. of steam per hour. The condenser consisted of corrugated-steel sheets spaced ¼ in. apart. The steam flowed inside the compartments, the air being circulated by a fan. In all about 43,000 sq. ft. of cooling surface was installed. A 22-in. vacuum was obtained with outside air at 42°F. With outside air at 113°F. no vacuum was obtained. The average was about 18 in. for the year. The fans took about 10 per cent. of the station output at full load.

Gebhardt gives for the value of heat transmitted from steam to air through ⅛-in. steel corrugated plates, 10 to 25 B.t.u. per hour per square foot per degree difference of temperature for air velocities of 500 to 4,000 ft. per minute.

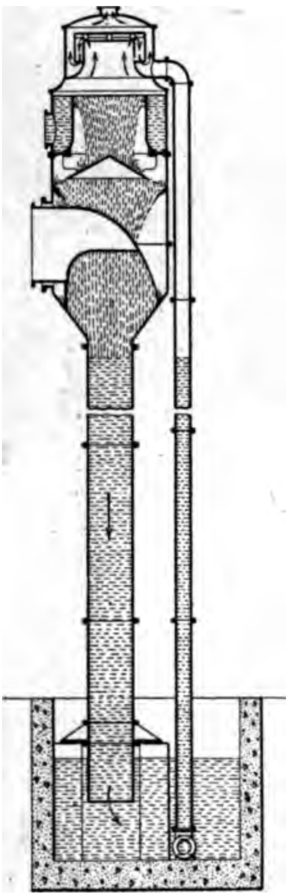


FIG. 51.—Section through Beyer barometric condenser.

Evaporative Condensers.—In this type of condenser the steam is condensed in plate or tube condensers by evaporating the cooling film of water. These condensers are common in refrigerating, brewery and similar industries. A slow air circulation is necessary to carry off the moisture-laden air. The heat transmission is quite good. With cast-iron pipes about 1 sq. ft. of surface is required per pound of steam per hour and vacuums up to 25 in. are readily maintained. With brass tubes and good circulation about 8 lb. may be condensed per hour per square foot of surface. The power for fans and pumps should not exceed 5 per cent. of the output (see paper by Oldham, *Proceedings* I.M.E., 1899).

FIG. 52.—Koerting eductor and multijet condensers.

Pennel in Kansas City built a number of evaporative condensers which are interesting. He secured a 25-in. vacuum with a cooling water evaporation equivalent in weight to 43 per cent. of the steam condensed (see *Power*, Jan. 12, 1909, page 128). He used a stack to secure the air circulation. Usually in evaporative condensers the weight of water evaporated is equal to from 50 to 90 per cent. of the weight of steam condensed. In another type built in Chicago a horizontal steel shell is provided with heads in which are expanded a large number of small brass tubes. The outlet end is provided with a cone leading to the exhaust fan which pulls the air through the tubes. The steam to be condensed is led between the tubes. Water is sprayed into the open ends of the tubes with the air.

Condenser Auxiliaries.—These may be classified as:

(*a*) Circulating pumps which supply the cooling water at a sufficient pressure to overcome friction and velocity heads.

(*b*) Hotwell or condensate pumps which serve to remove the condensate only from the condenser and deliver it against the atmospheric pressure.

(*c*) The wet air pump of which there are many varieties.

1. The air pump which not only removes the condensate and non-condensible vapors but also removes the cooling water which has been delivered to the condenser by atmospheric or other pressure.
2. The air pump which only handles the condensate and non-condensible vapors.
3. The air pump which is a combination condensate pump and wet and dry air pump. There are a number of designs of this type.

(*d*) The reciprocating dry air pump handling only the non-condensible vapors. This pump may be simple, compound or duplex.

(*e*) The water-jet air pump which is of many types.

1. The plain ejector type, single- or multiple-jet with separate pump.
2. The combined pump and ejector.
3. The pump and disk-jet type.
4. The pump and multiple-jet type.
5. The slug type with single or multiple diffuser.

Circulating Pumps.—The circulating pump supplies the cooling water to the condenser at a sufficient pressure to overcome the friction and velocity heads.

Plunger pumps, directly driven from the prime mover, were first used for this purpose and were quite satisfactory, but large and costly. The head is usually small, not over 20 ft. where the highest point in the circulating system does not exceed 30 ft. above the water level. The lower end of the discharge pipe can always be submerged and this condition maintained but the excess head must always be pumped against.

It was found that better results could be obtained by separating pump and prime mover and the first independent circulating pumps were plunger pumps of the direct type without flywheels.

To save space and weight the centrifugal circulating pump was introduced in marine practice and has now replaced all others for stationary practice as well. It may be driven by an engine, motor, or steam turbine either directly or through gearing or belting. For small sizes the best, cheapest and most satisfactory unit is the motor or steam-turbine driven unit directly coupled, the type of drive depending on the type of station. For large work the low-speed pump is driven by a high-speed motor through reduction gearing, and for very large work the pump may have two or three rotors in one casing. Pump efficiencies of 60 per cent. are usual and by careful design 70 per cent. can be attained.

Where a supply of water is available above the condenser level, circulating pumps may be dispensed with entirely and there are a few installations where the condenser may be worked without pumps of any kind. Such an installation may be seen at Rochester, N. Y., where ejector condensers without pumps are in use giving 27 in. of vacuum.

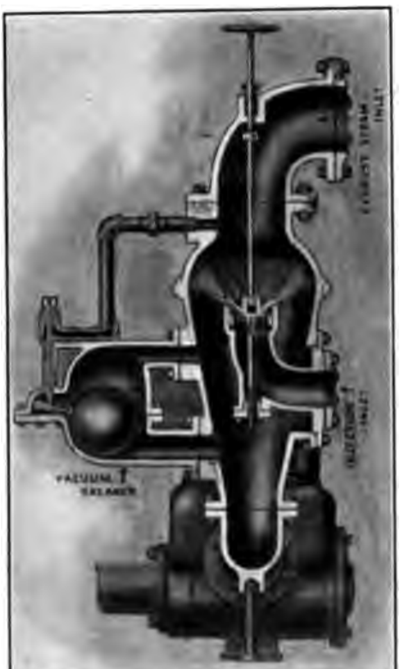


Fig. 53.—Blake-Knowles jet condenser.

Condensate Pumps.—Separate condensate pumps are a comparatively late innovation and are either of the direct-acting or centrifugal type. The direct-acting pumps have not been much used since the development of the centrifugal condensate pump. Centrifugal pumps or this service are never large, a 6-in. pump being sufficient for the largest unit. They were formerly built in two stages, but since the conditions under which they work have become better understood single-stage pumps have been employed. The head is rarely over 60 ft. of which 30 ft. may be due to the vacuum in the condenser and the rest friction and delivery head in the discharge line. These pumps should always be horizontal top discharge to avoid air pockets and the suction connection should be so designed as to be always submerged. The runners may be ventilated to the upper part of the hotwell, but if the suction piping is short and direct and the suction submerged this is not necessary.

The condensate and circulating pump may be combined in certain forms of mixed condensers with good results. The old jet condensers, Fig. 53, were examples of this type and the newer centrifugal jet, Fig. 57, has been quite successful. This type can be used only where the injection nozzle is less than 18 to 20 ft., above the water supply and means must be supplied for priming.

This arrangement is identical with the jet condenser of Watt and the pump may be termed an air pump. Watt's pump consisted of a cylinder provided with suction and discharge valves and also a third set of valves in the piston and was thus both a lift and force pump. Wet air pumps of similar design are common. One set of valves may be discarded. The common horizontal air pump is of this type. When two of these valve decks are absent the Bodmer, Edwards, or Brown pump results and many designs are on the market which give excellent service.

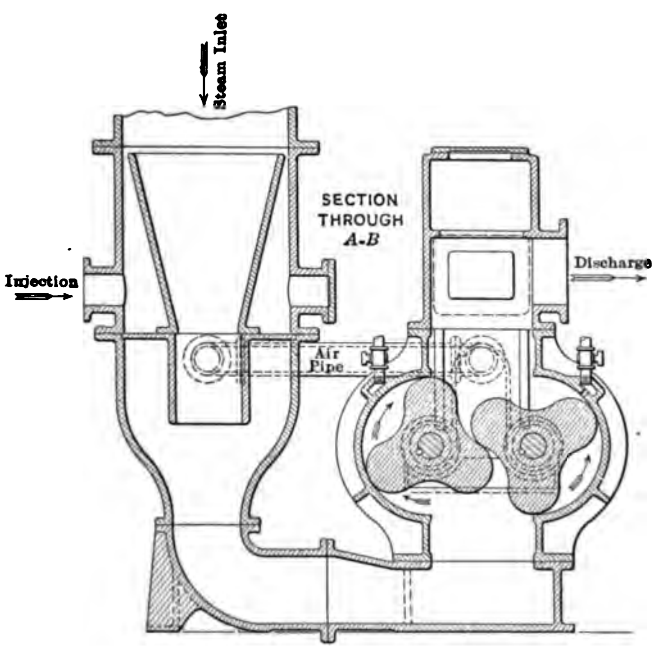


FIG. 54.—Connersville jet condenser.

These pumps are usually arranged for handling the condensate and non-condensible vapors and are known as wet air pumps.

The development of sugar manufacture showed the need for a pump to remove the non-condensible vapors and the air compressor was pressed into service for taking these gases at low pressures and compressing them to atmospheric pressure. It was found that the volumetric efficiency was very low and the Weiss bypass was applied to the air compressor. This bypass puts the two ends of the dry air cylinder in communication for a short time after the intake and discharge valves are shut and before the intake valves open equalizing the pressure on both sides of the piston and avoiding part of the loss due to expansion of the vapor in the clearance space. Practically all single-stage dry air pumps use some form of this bypass.

If the dry air pump works in two stages the bypass is not necessary. In these pumps the first stage usually compresses to about twice the

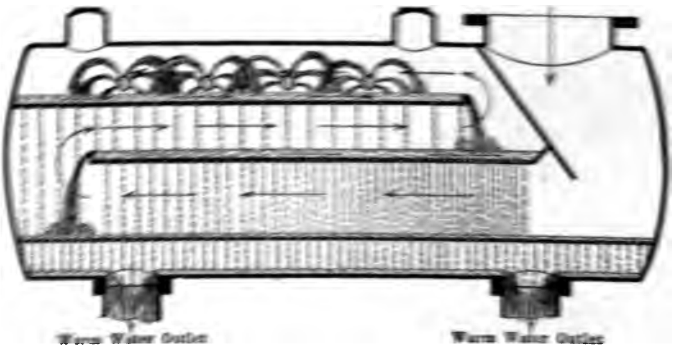


Fig. 55.—Counter-current "Rain-type" condenser.

absolute pressure, leaving the remaining portion to be done in the second stage. Not many of these pumps are in service, as they are both costly and bulky and the maintenance cost is high.

It has been found very difficult in practice to maintain reciprocating

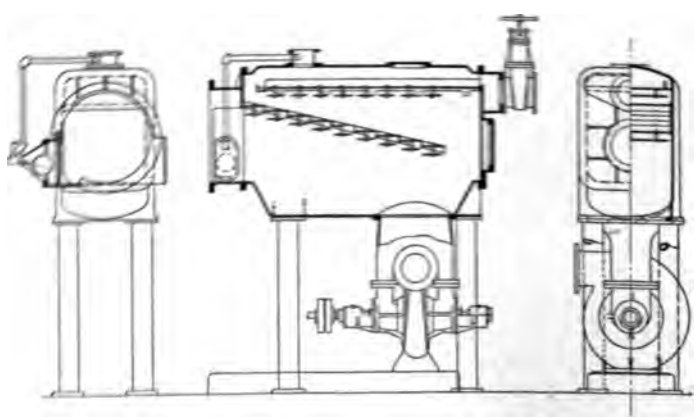


Fig. 56.—Wheeler rectangular jet condenser.

dry air pumps tight against leakage. This leakage has in some cases amounted to as much as 25 per cent. of the total weight of air to be handled which in connection with the bad volumetric efficiency is a serious handicap to the maintenance of a good vacuum.

Reciprocating dry air pumps are quite economical in the use of power.

The maximum power will be required at about 16 to 17 in. of vacuum and decreases as the vacuum improves.

Where reciprocating dry air pumps are used an air bell should be

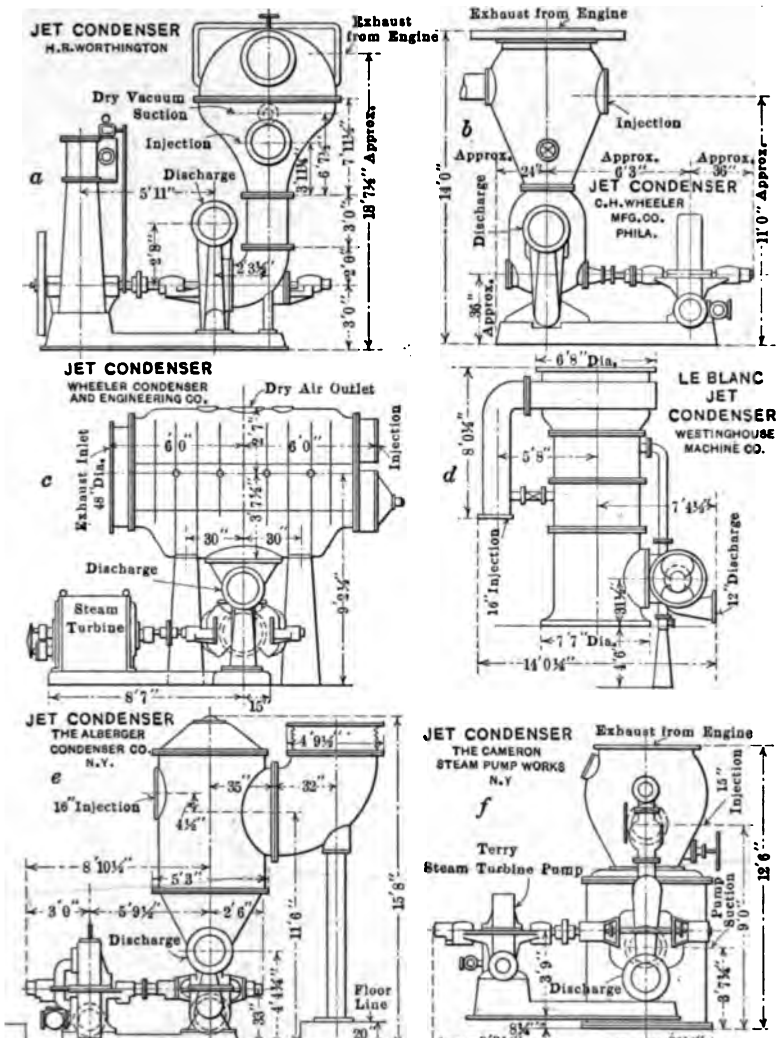


FIG. 57.—Six types of centrifugal jet condensers.

installed (see "Air in Surface Condensation," *Transactions* A.S.M.E., vol. 34). With this apparatus the total air and pump leakage can be tested and pumps and condenser shells maintained in good condition.

Water-jet air ejectors have long been known and the adaptation of these principles to condenser practice is well illustrated in the Koerting, Tomlinson and Bulkley ejector condensers. In these condensers the air is entrained with the condensate and cooling water and is carried away with the water.

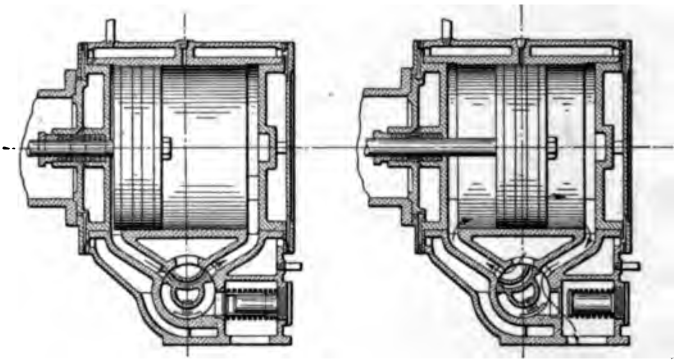


FIG. 58.—Dry-vacuum-pump air cylinder with Weiss bypass.

It was due to the Frenchman, Maurice Le Blanc, that these principles have been adapted to the abstraction of air from condensers. These pumps have three principal parts, a pump to impart pressure to the water, an entraining nozzle or nozzles, and a diffuser or diffusers, and differ only in the arrangement and location of their members.

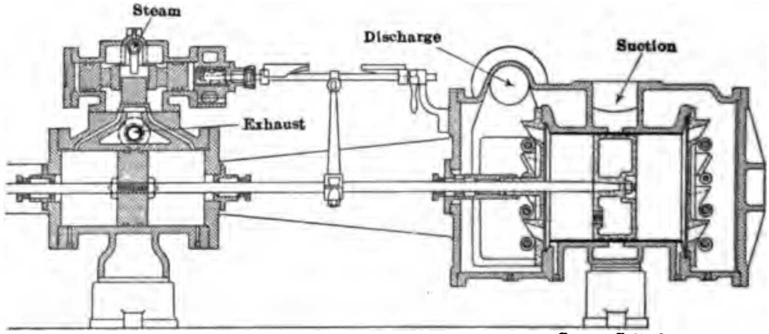


FIG. 59.—C. H. Wheeler suction valveless air pump.

In the simplest type the pump is separate from the entraining nozzle. The pressure water may be taken from the circulating pump or other source and the entraining nozzle and diffuser may be placed in the location most convenient to the air eduction nozzle. Such pumps are manufactured by the Worthington and Allis-Chalmers Co. in this country and by Willans & Robinson in England. The air nozzle and diffuser may even be placed inside the condenser as proposed by Josse and Gensecke and built by the L. Schwarz A. G. A multiple-nozzle pump of this type is made by Koerting.

Combination units are much more common and the plain cylindrical jet and diffuser combined with the centrifugal pump can be obtained. A better arrangement, however, is where the entraining nozzle entirely

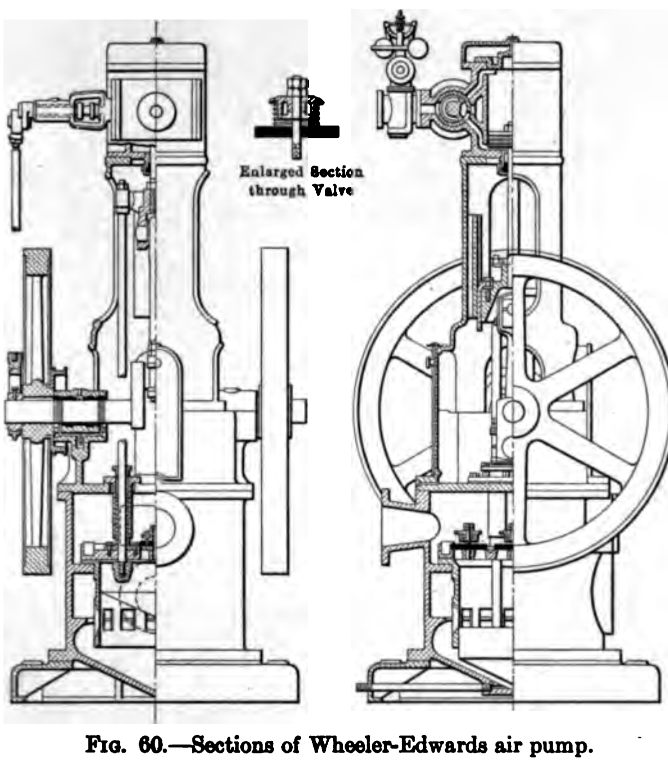


Fig. 60.—Sections of Wheeler-Edwards air pump.

surrounds the runner of the centrifugal pump as in the air pump invented by Kolb and improved by Pfleiderer. This pump is manufactured by Thyssen in Germany and by the C. H. Wheeler Mfg. Co. in this country. Following the same principle but with a widely differing detail is the Rees pump made by the Rees Roturbo Mfg. Co. in England and by the Manistee Iron Works in America.

The Kolb pump has a single continuous jet, a continuous entraining nozzle and a continuous diffuser, all radial. The Rees type has a single jet, a multiple-entraining nozzle and a multiple large-passage diffuser.

A very interesting and efficient pump built by the A. E. G. in Germany

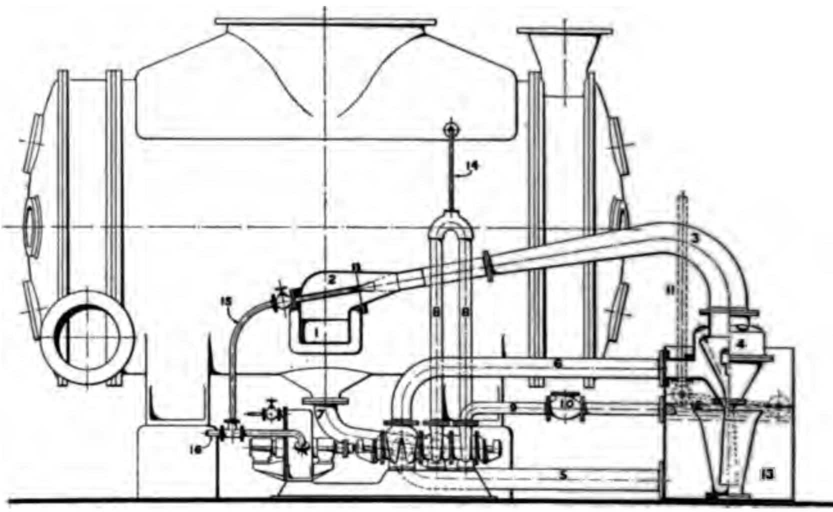


FIG. 61.—Condenser with kinetic air-pump system.

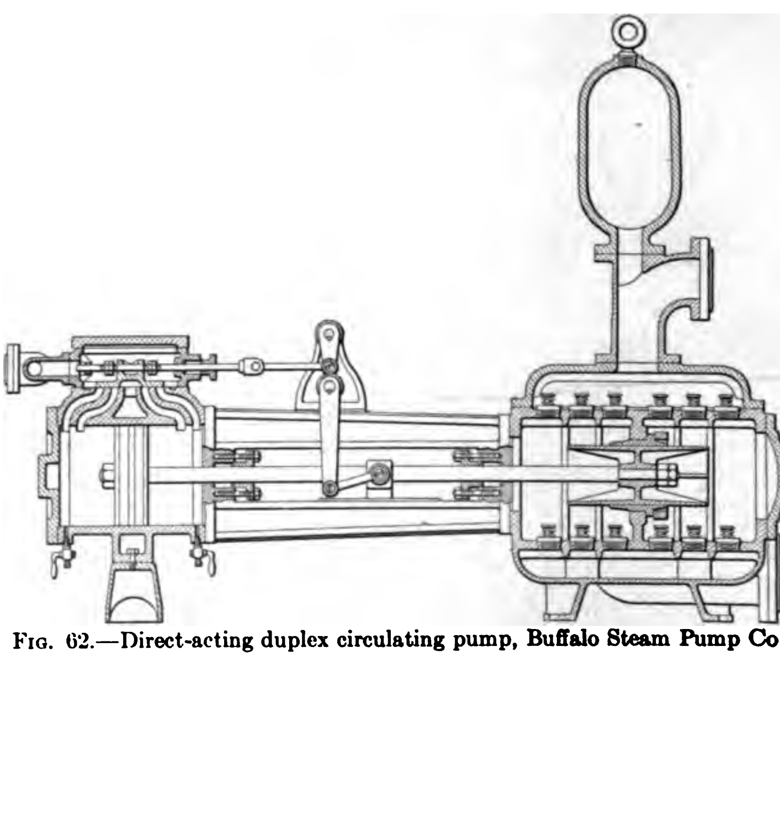


FIG. 62.—Direct-acting duplex circulating pump, Buffalo Steam Pump Co.

and by the Wheeler Cond. & Eng. Co. in this country is like both in principle, but differs in detail. The jet is broken, the entraining nozzle is simple and the diffusers are fewer and converging. This pump is

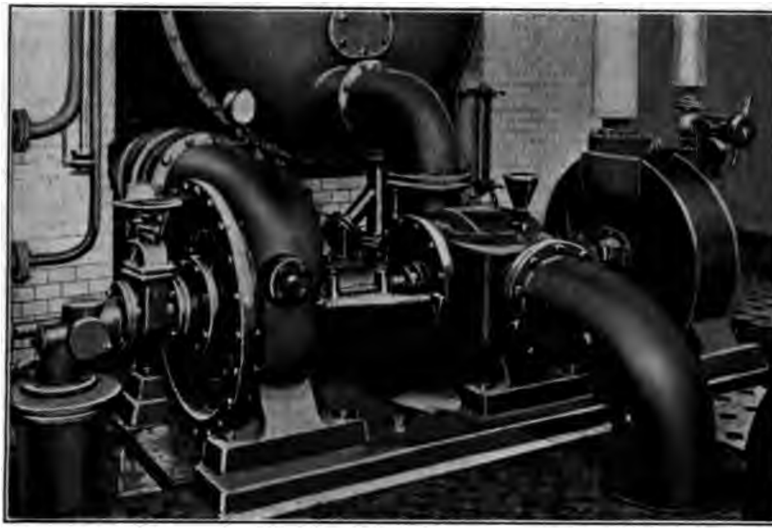


Fig. 63.—Condenser auxiliaries, A. E. G.

used to a large extent in Europe and has given very good service. This type may be termed the "slug" type and its action is exactly similar

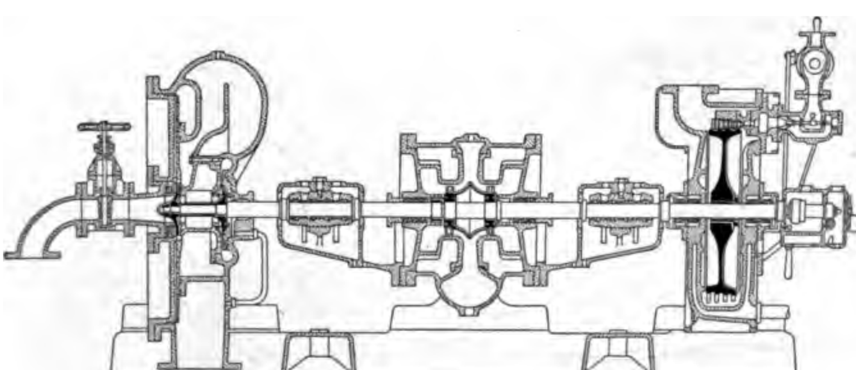


Fig. 64.—Section through condenser auxiliaries, Allgemeine Electricitäts Gesellschaft, Berlin.

to the Le Blanc pump which has only a single diffuser and a real entraining nozzle which is absent in the A. E. G. pump.

The Le Blanc pump is made in this country by the Westinghouse

Electric and Manufacturing Co. and by most all builders abroad. A particularly good pump of this type is made by Weir of Glasgow. The Le Blanc pump as made by the Westinghouse company is particularly adaptable to mixed condensation and many units of this type are in service. The Rees pump is also used for this purpose.

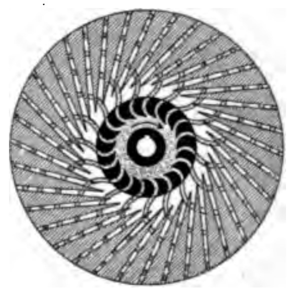


FIG. 65.—Section through diffuser A. E. G. air pump.

The Kinetic System.—A development of the Parson's augmenter and the water-jet pump is the kinetic air-pump system of Richardson, Westgarth & Co. Here the air is exhausted from the condenser by the steam jet (Parson's augmenter) and delivered to the kinetic ejector which takes its water and discharges it into the feed tank. The kinetic ejector

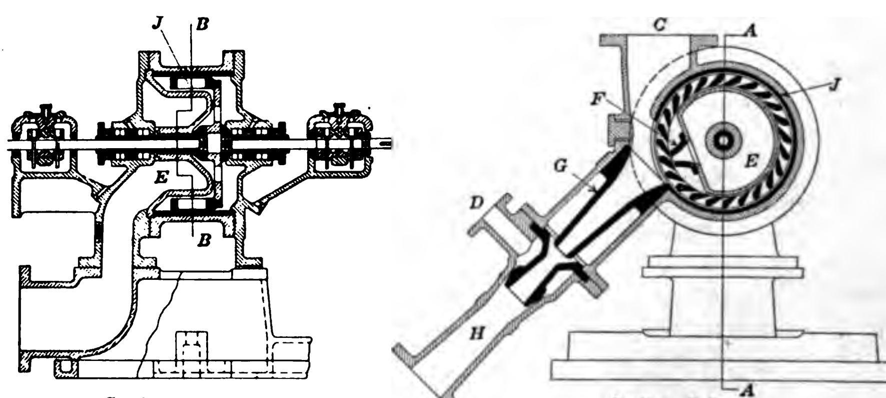


FIG. 66.—Sections of Westinghouse-Le Blanc air pump.

is the condenser for the augmenter and also warms the feed water as well. This system has many advantages and was installed in connection with the 25,000-kw. Parsons turbine at the Fiske Street Station of the Commonwealth Edison Co.

It should be remembered that the vacuum obtained by the use of water-jet pumps is dependent on the temperature of the water used in the jet. Grunewald, in *Z.V.D.I.*, Dec. 7, 1912, has presented a most complete set of tests on these pumps measuring the air with nozzles and his curves are worthy of careful study. The power used is much larger than with the reciprocating pumps because of the large amount of water handled. The compression is nearly isothermal and even large amounts of air make little difference in the power.

The chief advantage of the water-jet pump is the absence of repairs and attendance, as it is usually driven by a steam turbine or motor. The extra power is a small percentage of the whole, although four or five times that used by a reciprocating pump. The water for the jet must be clean or very carefully strained and in some installations it has been especially cooled to secure better vacuums.

The chief disadvantage of this type is the difficulty of using the air bell for determining the air leakage into the system, as the pump diffuser usually discharges directly to the discharge tunnel. Discharge wells may be provided or a dry air pump installed with piping to each condenser for testing purposes.

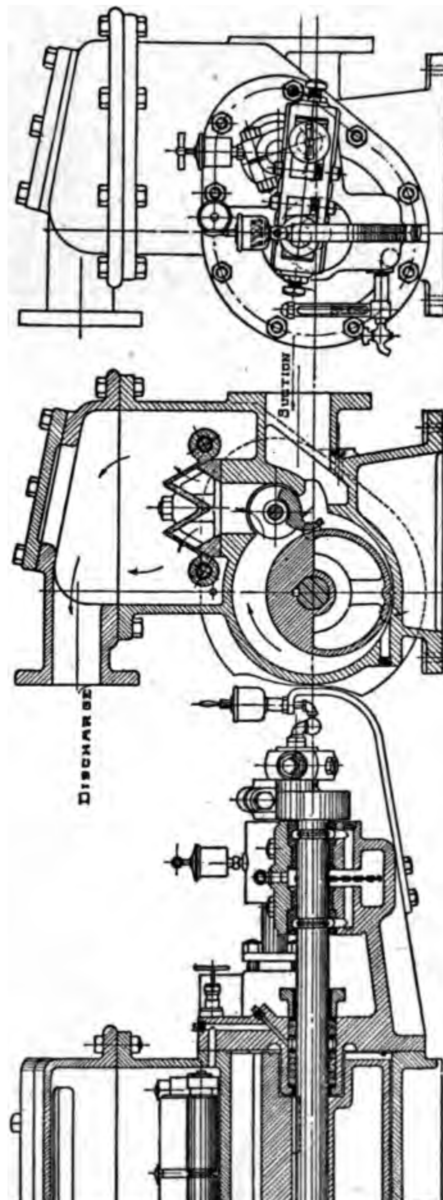


Fig. 67.—Rotrex dry air pump, C. H. Wheeler Cond. & Pump Co.

Hotwells.—Surface condensers should be provided with a hotwell so shaped as to retain sufficient condensate to drown the condensate pump. The delivery of this pump may be controlled by a float.

Where duplex or triplex pumps of the Edwards type are used the pipes leading to the pumps are usually sufficient.

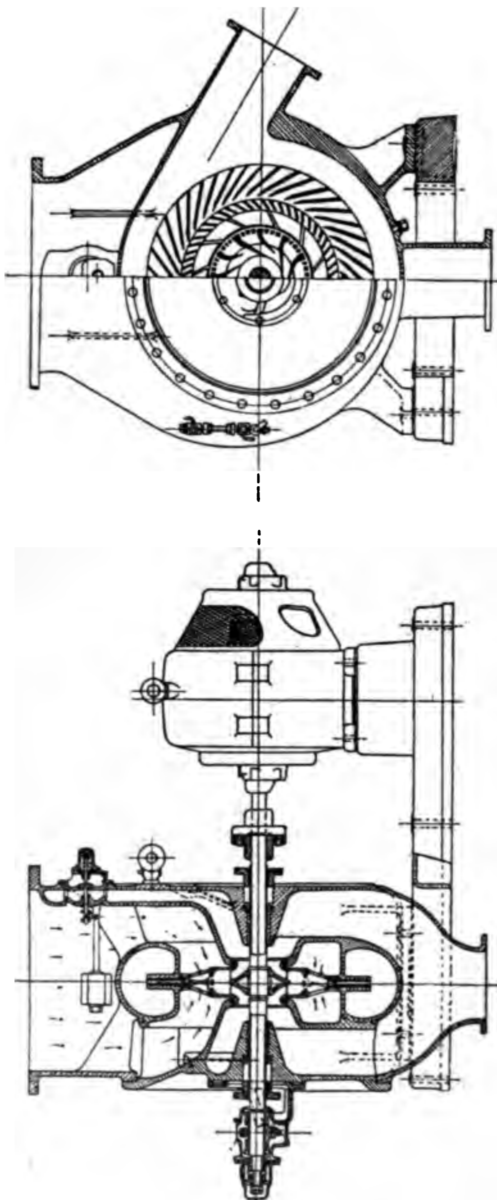


Fig. 68.—Rees roturbo rotary jet condenser.

Priming.—With many types of condensers priming is necessary to start the system. With reciprocating dry air pumps the pump may be started and a vacuum created in the condenser. A pipe connection with valve between the water boxes and the condenser will provide for the exhaustion of the circulating system and the water will rise to the circulating pump which can then be started and the condenser is ready for work. A steam ejector applied to the water box will perform the same service. Water jet vacuum pumps are best primed from the house service lines and after starting can exhaust the system and raise the water to the circulating pumps.

Expansion Joints.—An expansion joint is usually fitted between the condenser and prime mover, and varies in size from the 3 to 6-in. diameter copper joints on very small units to the large 10 by 15-ft. joints on a 30,000-kw. unit. Corrugated copper is largely used for joints of this kind, but the best joint is of the Baragwanath type made of steel plate which may be welded to the distance pieces.

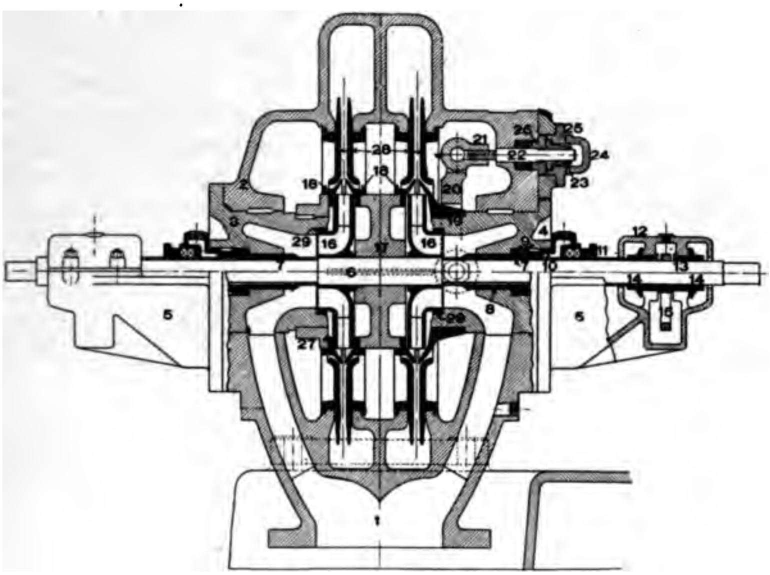


FIG. 69.—Thyssen vacuum pump.

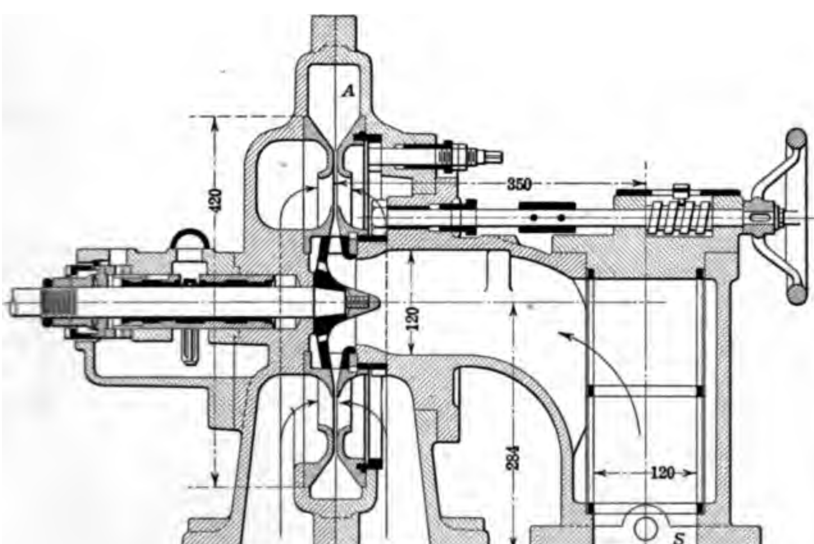


FIG. 70.—Kolb jet condenser and air pump.

This joint has been omitted of late, on very large units and the condenser hung from the steel foundation. In this case smaller expansion joints are required in the circulating, air and hotwell connections.

Power Required for Condenser Pumps.—The power required by the air and water pumps connected with condensing apparatus is approximately 2 to 5 per cent. of the indicated horsepower of the main units. J. R. Bibbins (*Power*, February, 1905) reports for test conditions the following figures for a 3,000-kw. plant.

Indicated horsepower of auxiliaries	151	200	238	260	291	294	457	589
Per cent. total power used	4.69	3.51	3.22	3.22	3.08	2.97	2.80	2.47
Per cent. for air pump	1.63	1.36	1.27	1.21	1.19	1.09	0.95	0.85
Per cent. for water pump	3.07	2.14	1.95	2.00	1.90	1.89	1.85	1.52

For a 2,000-kw. turbine at the St. Louis Exposition at full load, the power input to auxiliaries was 7 per cent. The 7 per cent. includes transformer and motor losses.

Test of Auxiliaries of 5,000-Kw. Unit of Boston Edison Co.

Kilowatts on turbine	2,713	3,410	4,756
Vacuum	28.4	28.7	28.6
Barometer	29.53	29.95	29.96
	Horsepower developed		
Boiler-feed pump	13.9	23.7	27.4
Circulating pump	69.1	69.1	69.1
Dry vacuum pump	24.3	23.3	23.8
Step-bearing pump	6.4	5.8	5.6
Wet vacuum pump	8.6	9.2	9.8
	122.3	131.0	136.7
Per cent. power of auxiliaries to power of turbine	3.4	2.9	2.1
Per cent. water used by auxiliaries to that used by turbine	8.4	7.4	5.7

Finch in the *London Electrician*, Mar. 22, 1912, gives the following figures of the use of power for auxiliaries in English power stations.

Location	Main plant kw. rating	Power used for auxiliaries per cent. of main plant rating						
		Feed pumps	Stoker economiser	Fans	Con-denser aux.	Coal and ash	Miscel-laneous	Total, per cent.
Chelsea, London	48,000	0.5	0.5	...	1.9	0.7	1.1	4.7
Carville, Newcastle	35,000	1.2	0.4	3.3	4.1	0.3	0.5	9.8
Greenwich, London	34,000	1.2	0.35	...	3.4	0.7	0.45	6.1
Port Dundas, Glasgow	22,400	1.1	0.3	2.1	3.7	0.5	0.3	8.0
Dunston, Newcastle	19,000	1.0	0.25	1.4	2.6	0.3	0.7	6.25
Standreens, Glasgow	16,400	0.9	0.3	0.8	7.8	0.5	0.2	10.5
Brighton	10,200	1.2	0.35	4.5	2.4	0.275	0.275	9.0
Stepney	6,000	1.1	0.35	1.2	4.9	0.65	1.4	9.6
Newcastle & District Co.	5,450	0.8	0.35	2.2	4.8	0.1	0.45	8.7
Cambridge	1,880	2.1	0.1	0.1	3.35	.....	0.55	6.2
Alnwick	140	10.7	....	...	.....	.....	.....	10.7
Morpeth	60	11.0	....	...	10.0	.....	.....	21.0

He also gives figures for two German stations.

Hamburg Overhead	7,900	3.2	0.3	....	2.5	0.0	0.5	7.4
Markische (Berlin)	7,200	2.2	0.15	1.25	2.5	0.7	0.2	7.0

Cost of Condensers.—The average cost of surface condensers with pumps is indicated below:

Engine hp. simple	Engine hp. compound	Condensers cost	Erection cost	Total cost
40	50	$260	$90	$350
60	100	380	125	505
100	120	490	160	650
120	150	540	175	715
150	200	610	200	810
200	250	670	230	900
250	325	775	250	1,025
320	425	975	325	1,300

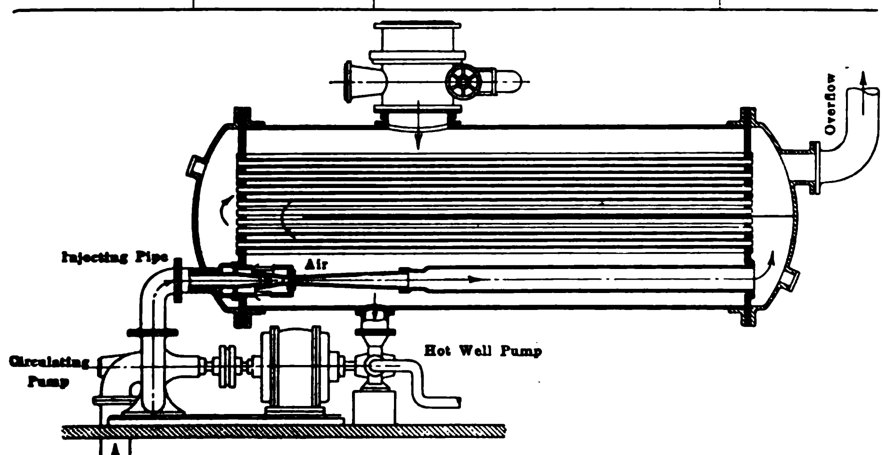


FIG. 71.—Ejector air pump inside condenser, L Schwarz A. G., Dortmund, Westphalia.

From this table it will be noted that the cost of erection is about 33 per cent. of initial cost.

The circulating and air pumps are included in the above table.

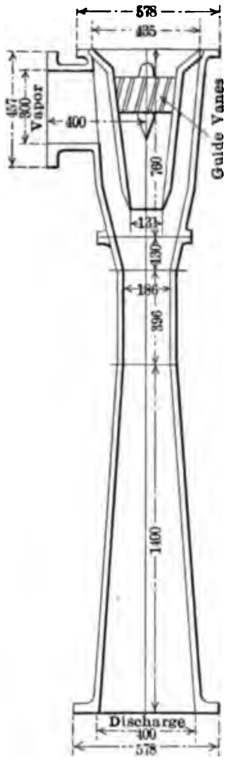


FIG. 72.—Willans-Müller ejector air pump 8,500 kw. turbine, Sheffield, Eng.

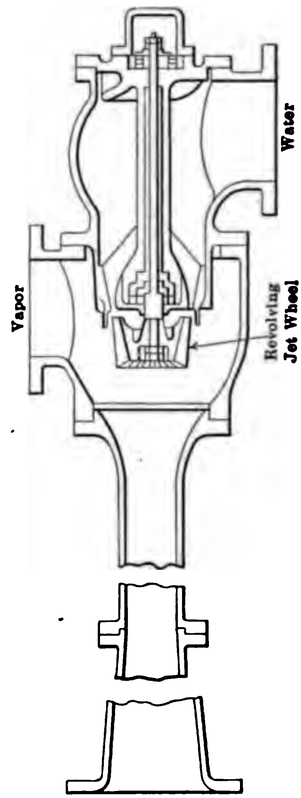


FIG. 73.—Hydraulic vacuum pump, H. R. Worthington, Inc.

Reducing the above table to cost per horsepower for convenience in applying to future estimates the following figures result.

Engine hp. simple	Cost per hp.	Engine hp. compound	Cost per hp.
40	$6.50	50	$5.20
60	6.43	100	3.80
100	4.90	120	4.08
120	4.50	150	3.60
150	4.07	200	3.05
200	3.35	250	2.68
250	3.10	325	2.38
320	3.04	425	2.29

For installations from 500 kw. to 6,000 kw. the cost of various types of condensers completely installed is reported as:

Type	Cost per kw.
Siphon, without air pump	$2.00–$3.00
Jet	3.00– 4.50
Barometric with dry air pump	4.00– 6.00
Surface for 26-in. vacuum	3.50– 5.00
Surface for high vacuum	6.00–10.00

Cost of Individual Condenser Installations.—A condenser and direct-acting pump for a 1,200-hp. installation cost, exclusive of erecting, $2.50 per horsepower.

A 1,550-sq. ft. surface condenser for a 750-kw. engine (2 sq. ft. per kilowatt); 26-in. vacuum; direct-acting pump underneath condenser and centrifugal circulating pump and engine at end of condenser; cost f.o.b. factory $2,650 or $3.50 per kw.

The price of ordinary jet condensers with connecting pipes may be taken as from $750 to $2,000 for engines varying from 300 to 2,000 hp.

Formulæ for Cost of Condensers.—An approximate formula that is sometimes used for determining the cost of jet condensers is,

$$\text{Cost in dollars} = 500 + 1.0 \times \text{hp.}.$$

Potter (*Power*, Dec. 30, 1913) gives:

Type	Capacity	Cost $
Barometric (28-in. vac.)	Up to 30,000 lb. steam per hour.	$1,055 + 0.112 \times$ (lb. steam cond. hr.)
Jet (28-in. vac.)	Up to 30,000 lb. steam per hour.	$1,176 + 0.1138 \times$ (lb. steam cond. hr.)
Surface (28-in. vac.)	Up to 35,000 lb. steam per hour.	$1,630 + 0.2038 \times$ (lb. steam cond. hr.)
Surface (26-in. vac.)	Up to 30,000 lb. steam per hour.	$413 + 0.1015 \times$ (lb. steam cond. hr.)

Twenty-eight-inch vacuum surface condensers with pumps cost from $1 to $1.80 per square foot of surface depending on price of copper, or from $150 to $250 per 1,000 lb. of steam condensed per hour.

Cooling Ponds.—Where water for condensing is scarce it is found economical to store it in reservoirs where it is cooled by evaporation. Under the conditions prevailing in northeastern United States it has been found that a surface of 250 sq. ft. is sufficient to cool the condensing water required for a boiler horsepower (34.5 lb.) at 26-in. vacuum. In countries where the evaporating coefficient is higher, smaller surfaces may be used (see Ruggles, *Transactions* A.S.M.E., vol. 34, page 561).

Spray Ponds.—Where the area available for water storage is smaller recourse must be had to the spray system. Here the warm condensing water is sprayed into the air above the smaller pond; a portion evaporates, cooling the remaining portion which falls to the tank to be used again. The sides of the tank are extended to prevent the spray from being carried away by the air currents. About 4 sq. ft. of surface are required for a boiler horsepower of steam condensed at 26-in. vacuum. Such a plant is installed at the power house of the Philadelphia Rapid Transit Co. at Second Street and Wyoming Avenue. It is even possible in small plants to have the spray pond on the roof of the power house.

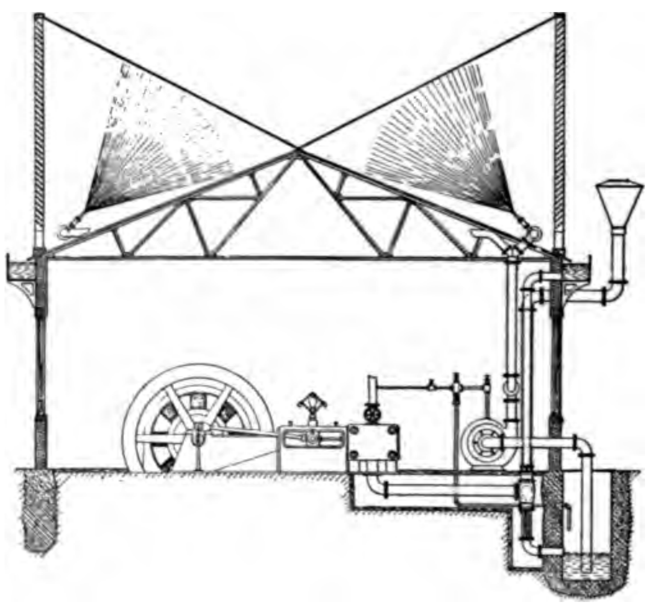


FIG. 74.—Cooling pond, Koerting spray system.

These ponds should not be used in crowded localities as the spray of water is likely to cause a nuisance. The loss of water varies between the amount of boiler feed and twice the amount.

Cooling Towers.—Where the available space is much restricted the cooling tower will cool the circulating water to the required temperature. The cooling tower consists of a chimney-like structure which may be provided with trays, curtains, baffles or gratings. The hot circulating water is pumped to the top of the tower where it is sprayed over the gratings and allowed to trickle downward to the cold well, exposed to the ascending current of air. Natural draft towers depend on the temperature of the water to produce the air circulation. In the forced draft towers this circulation is accelerated by a fan. There are many designs

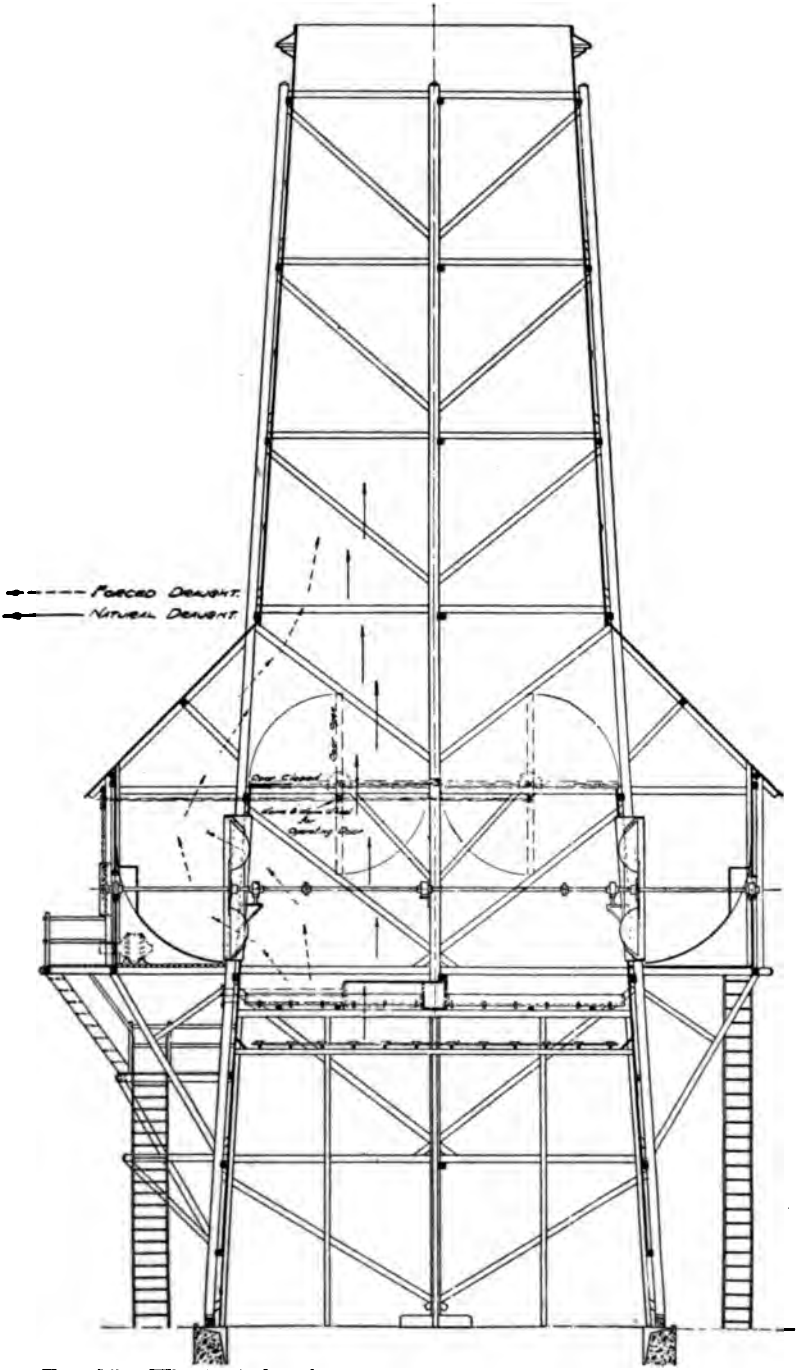


Fig. 75.—Wheeler induced-natural-draft cooling tower, Waco, Texas.

of these towers on the market but when well designed there is little difference in their efficiency. The loss of water is usually from 10 per cent. to 20 per cent. With the forced draft towers the fans may use from 2 per cent. to 4 per cent. of the output of the main plant.

Cooling towers and air condensers are quite efficient and can usually cool to within 2° to 5° of the wet bulb air temperature. They are rather bulky and costly but are quite advantageous when the limit of cooling is sufficiently low and their use is demanded by circumstances. It should be remembered that a decrease in the lower temperature of about 20° corresponds to an increase in capacity of tower of 100 per cent.

FIG. 76.—Worthington forced-draft cooling tower.

Cost of Cooling Towers.—Assuming 27-in. vacuum, 70° air and 70 per cent. humidity, forced-draft cooling towers cost about $250 per 1,000 lb. of steam condensed. For 24-in. vacuum the cost may be one-half this. Natural-draft towers cost about one-half to one-third as much as the forced-draft towers. As a general rule cooling towers do not pay when the pumping head exceeds 75 ft.

Cost of Water.—Water for condensing purposes, boiler feed, etc., may be an expensive item in city power plants. The usual rate for water for manufacturing purposes in cities is 40 cts. per 1,000 cu. ft., or about 5.35 cts. per 1,000 gal. In New York City this rate is $1 per 1,000 cu. ft.

PROBLEMS

22. Determine the proper design of a condenser for a 1,000-hp. compound condensing Corliss engine for a nozzle vacuum of 25.8 in. referred to a 30-in. barometer and a 65°F. circulating water. ¾-in. aluminum bronze tubes.

23. If a new surface condenser is to be installed for the engine of problem 11, how many square feet of cooling surface will be required? What would you expect to pay for the condenser and pumps, f.o.b. factory?

24. The vacuum for a 750-kw. turbo-generator installation is to be furnished by a jet condenser. Estimate the pounds of cooling water that will be necessary to give a vacuum of 28 in. (30-in. barometer), the injection water temperature being 68°F.

Could this vacuum be maintained with injection water at 85°F? If so, how much cooling water would be required? Estimate the cost of the condenser. A bidder offers a condenser with a 12-in. exhaust pipe diameter, a 7-in. tail pipe and a 6-in. injection pipe. Would you consider this condenser adequate?

25. Let it be desired to design a condenser for turbine service, 30,000 lb. of steam to be condensed per hour, circulating water 70° with a vacuum in the nozzle of 28.4 in. referred to a 30-in. barometer. Clean salt water for condensing and admiralty tubes 1 in. diam. No. 18 BWG. Determine:

(*a*) Pounds circulating water per hour.
(*b*) Square feet of tube surface.
(*c*) Number and length of tubes.
(*d*) Size of water connections.

26. Determine the data called for in problem 25, for a condenser to handle the same amount of steam, the main unit to be in this case a reciprocating engine operating under a vacuum at the condenser nozzle of 26 in. referred to a 30-in. barometer. Circulating water inlet temperature and tubing to be the same as in problem 25.

27. Determine the relative approximate cost of the following types of engines for a unit of 225 hp.

A. Simple, high-speed, non-condensing.
B. Compound, high-speed, non-condensing.
C. Compound, high-speed, condensing.
D. Simple Corliss, non-condensing.
E. Compound Corliss, non-condensing.

Determine for each type:

(*a*) Cost of engine.
(*b*) Cost of condenser, air and circulating pumps.
(*c*) Cost of foundation, erecting, etc.
(*d*) Total cost of engine, etc., erected.
(*e*) Probable pounds of steam per hour at rated load.
(*f*) Boiler horsepower required for each type.

28. Same as problem 27 substituting a 450-hp. unit.

29. Find the same data as in problem 27 for an installation 1,000 hp. divided as follows:

1. Two 500-hp. simple non-condensing Corliss engines.
2. One 750-hp. compound condensing Corliss engine.
   One 250-hp. compound high-speed non-condensing engine.
3. One 600-hp. simple condensing Corliss engine.
   One 300-hp. compound condensing high-speed engine.
   One 100-hp. simple non-condensing high-speed engine.

30. The manager of a small factory is considering the installation of a direct-connected steam unit of 75-kw. capacity. Two types of engines are under consideration, viz., (*A*) a simple, Corliss, non-condensing; and (*B*) a compound, high-speed, condensing. He desires to know:

1. (*a*) The cost of engine *A* erected.
   (*b*) The cost of engine *B* with condenser erected.
2. Size of boiler required.
   (*a*) With type *A* engine.
   (*b*) With type *B* engine.
3. If all water used is wasted, the cost of water per 308-day year, 11 hr. per day, with full load on the engines.
   (*a*) With type *A* engine.
   (*b*) With type *B* engine.

31. Determine the items listed below for the following direct-connected A.C. generating units, each rated to deliver 600-kw. load.

A. Compound, high-speed, condensing.
B. Compound, Corliss, condensing.
C. Condensing turbine.

Find for each type:
- (*a*) Cost of generating unit.
- (*b*) Cost of condensing equipment.
- (*c*) Cost of foundations, erecting, etc.
- (*d*) Total of the above.
- (*e*) Boiler horsepower required at rated load.
- (*f*) With steam costing 22 cts. per 10,00 lb., the cost of steam per 24 hr. if
  - (*a*) Continuous full load is carried;
  - (*b*) Full load is carried for 10 hr. and 40 per cent. of full load is carried for 14 hr.

CHAPTER VI

THE STEAM BOILER

There are two conditions governing boiler construction:

(*a*) Economy in liberation of energy from the fuel and in generation of pressure in the boiler.

(*b*) Safety in storage of heat energy.

Economy may be divided into:

1. Economy in first cost of boiler and appurtenances and in the setting which it may require.

2. In the combustion of the fuel or in the number of foot-pounds of energy derived from the heat units resident in the fuel.

3. In maintenance and repairs, in which would be included depreciation from use and age.

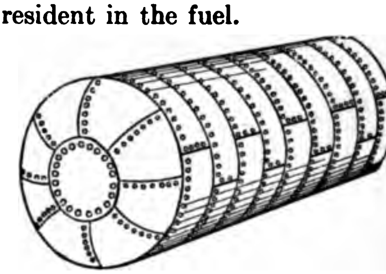


FIG. 77.—Egg-ended boiler.

The prime requisites of a boiler are, first, safety; and second, capacity when being driven.

To meet the condition of, "safety first" some authorities on boiler operation would impose the following requirements:

1. Never use anything but water-tube boilers.
2. Never use drums greater than 36 in. in diameter.
3. Always use two valves on all connections to the boiler.

Types of Boilers.—Formerly boilers were frequently classified as externally fired and internally fired, but the commercial classifications today are fire-tube and water-tube.

Fire-tube boilers include:

(*a*) Horizontal cylindrical boilers.
(*b*) Cylindrical-flue boilers.
(*c*) Return-tubular boilers.
(*d*) Locomotive-type boilers.
(*e*) Vertical fire-tube boilers.

Water-tube boilers include the many types of:

(*f*) Sectional boilers.
(*g*) Coil boilers.
(*h*) Water-tube boilers.

The plain cylindrical boiler was the first historically. It must be kept more than half full of water as is also true of all externally fired boilers. This type is now used little outside of old plants, in blast-furnace practice and in sugar manufacture. Such boilers are very simple, carry a large volume of water, steam very steadily after the large mass of water is once heated and require little attention. They are, however, wasteful of fuel and slow in getting up steam as the heating surface is small in proportion to the mass of water.

Cylindrical-flue Boilers.—The introduction of one or more flues (usually 1, 2, 3 or 5) into the plain cylindrical boiler added materially to the heating surface, thus shortening the boiler length and at the same time extracting the heat from the gases.

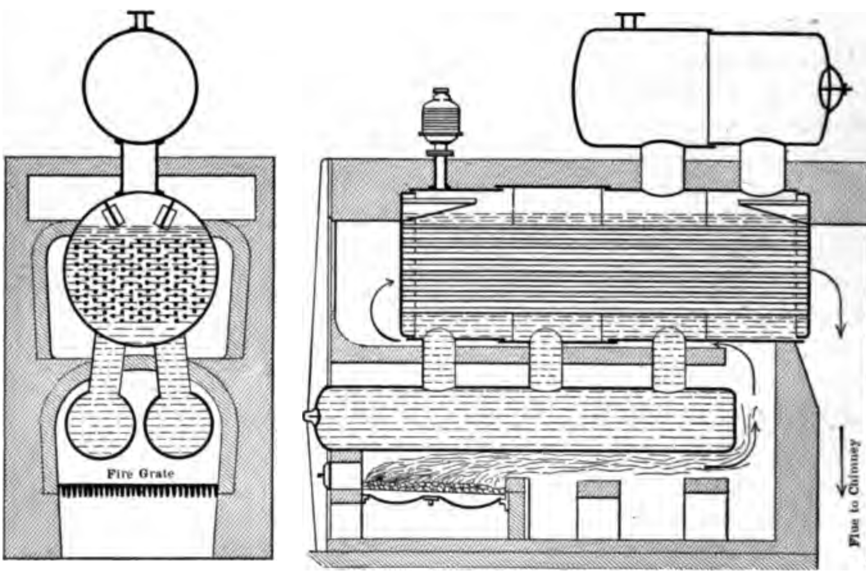


Fig. 78.—Elephant boiler.

The introduction of these large flues led to the introduction of the fire grate into the front end of the flue giving an internally fired boiler. The single-flue internally fired boiler is known as the Cornish type. The two-flue internally fired is called the Lancashire type.

Return-tubular Boilers.—Flues are tubes over 6 in. in diameter. By a reduction in the diameter and an increase in the number of tubes inserted the tubular boiler was developed. The heating surface is thus greatly increased and the efficiency of the boiler improved. "Getting up steam" is also much easier than with the plain cylindrical boiler. This is the most generally used type of fire-tube boiler.

In the ordinary fire-tube boiler the tubes are from 2 to 4 in. in diameter and their usual length is 12 or 16 ft. Locomotive boilers, dry-back and Scotch marine boilers and vertical fire-tube boilers are but special forms of tubular boilers.

Water-tube Boilers.—Water-tube boilers consist of a number of generating vessels so joined together that the steam formed in all of these separate units, or sections, is delivered from a common disengagement surface into a common steam space. Such boilers are also known as sectional boilers and coil boilers according to the method of constructing and combining the units.

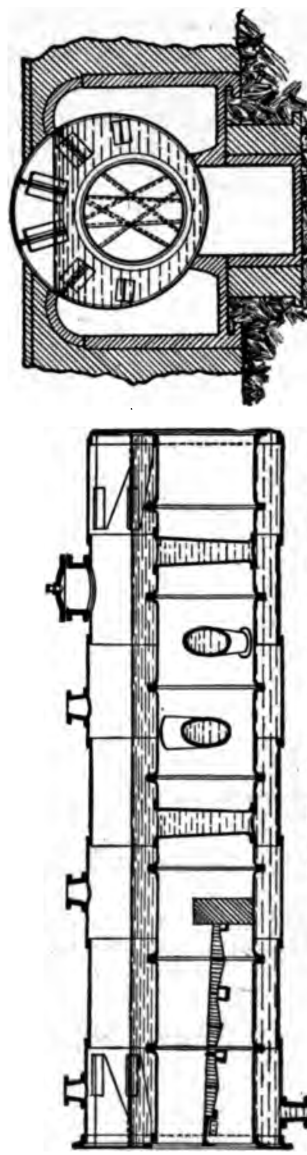


Fig. 79.—Longitudinal and cross-section of Cornish boiler.

Advantages of Sectional Type.—

1. Each section is far safer against rupture than a large shell of the same thickness.
2. Rupture or failure of one section does not usually cause failure of the whole structure.
3. Tubes can be thinner than would be possible with shells, thus giving a lighter weight for a given evaporative capacity.
4. Easy portable.
5. Repairs and renewals easy, rapid and cheap.
6. Can be driven further above normal rating than shell boilers of most types.

Disadvantages of Sectional Type.—

1. Units must be connected steam- and water-tight. Unequal expansion and contraetion tend to loosen joints.
2. Question whether the circulation is positive.
3. Failure of a tube necessitates shut down for repairs. Can plug a fire-tube.
4. Shapes that cannot be properly cleaned or inspected are bad.
5. Gases are liable to pass too rapidly through the system and to leave at too high temperature.
6. Workmanship and parts of sectional boiler make it costly.

A brief summary of the advantages and disadvantages of the principal types of boilers in commercial use at the present time is as follows:

Cylindrical-flue Boilers.—

Advantages.—

1. Simple in construction.
2. Easily cleaned and examined.
3. Large steam space.
4. Not liable to prime.

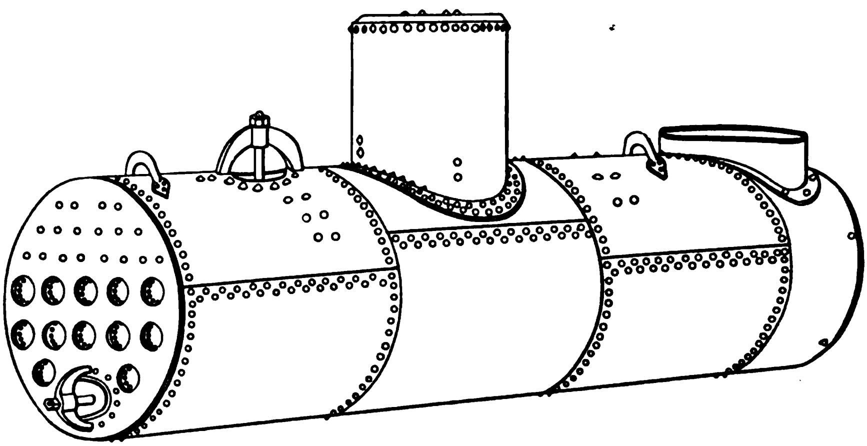


FIG. 80.—Horizontal-flue boiler.

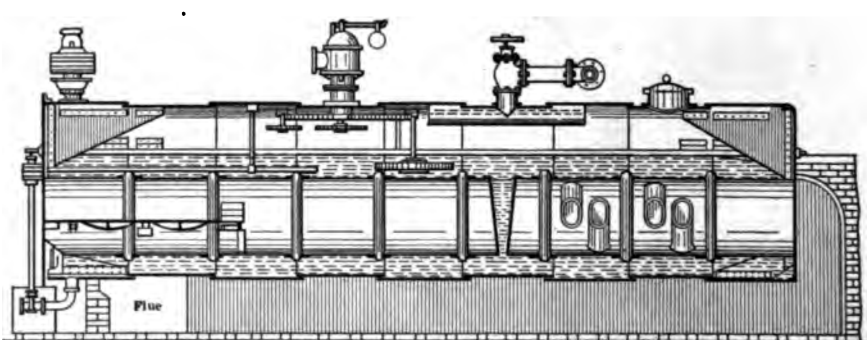


FIG. 81.—Section of Lancashire boiler.

Disadvantages.—

1. Slow steaming.
2. Liability to leak from unequal expansion.
3. Large floor space required.
4. Specially skilled men required for repairs.
5. Reduction in pressure necessary after a time.

Return-tubular Boilers.—

Advantages.—

1. Small floor space required.
2. Quick steaming.
3. Ruptured tubes easily replaced.

Disadvantages.—

1. Small steam space, hence pressure liable to fluctuate.
2. Not easily cleaned or examined.
3. Liable to leak at end of tubes, at stays and at corners of firebox.
4. Reduction of pressure necessary in time.

Water-tube Boilers.—

Advantages.—

1. Rapid steaming.
2. Relatively small danger from explosion.
3. Small floor space required.
4. Repairs easily made.
5. Respond readily to changes in steam demand.
6. Freedom of expansion.
7. Positive circulation.
8. Adaptability to high pressures.
9. Ease of installation.
10. Elasticity of design.
11. Large overload capacity.
12. Reduction of pressure not necessary.

Disadvantages.—

1. Small steam space.
2. Small water reserve.
3. Large number of parts.
4. Tubes difficult of access in some types.

As the majority of the fire-tube boilers already considered are horizontal boilers, the special advantages and disadvantages of the vertical fire-tube types are presented.

Advantages.—

1. Light and portable.
2. Requires no setting.
3. Rapid steamer.
4. Takes little floor space.
5. Upright flow of hot gases is the natural one.
6. Simplicity of staying, etc., makes it cheap.

Disadvantages.—

1. Circulation apt to be defective.
2. Troublesome to get a dome or large steam space.
3. Upper ends of tubes not water-cooled in many types and they necessarily get very hot, expanding and causing leakage.
4. Not easy to clean or inspect.
5. Hold little water. Pass to dangerous pressure quickly.
6. Strains in certain portions are exceedingly difficult to calculate.

The special features of internally fired boilers are:

Advantages.—

1. Economy. Little heat lost by radiation.
2. On account of 1, the fire rooms should be more comfortable.
3. Surface is efficient for evaporation, making boilers compact.
4. Furnace being internal, the boiler requires either no setting or one of the simplest description.
5. No cold air infiltrates through the setting.

Disadvantages.—

1. Internal firebox under pressure tending to collapse it inward makes it costly.
2. Efficiency of heating surface keeps temperature of combustion down and makes smoky products of combustion.
3. Rapid steaming capacity apt to cause sudden increase in pressure, even to the danger point.
4. Many types hard to clean and inspect.
5. Circulation not always satisfactory.

Dependability of the Different Boiler Types.—1. All commercial boilers will give, under the best conditions, practically the same economy in the evaporation of water; a boiler and furnace efficiency of 70 to 80 per cent. having been obtained with nearly every type of boiler which has been tested carefully under good conditions.

2. The life of all commercial boilers is at least from 15 to 20 years, probably much longer.

3. The only difference between the various types of boilers for power-station purposes is the space which they occupy, and the maximum rating at which they can be run commercially in everyday service. These considerations limit the choice of a boiler for the power station to a very few types.

Materials.—The requisites of boiler materials are sufficient tensile strength and a ductility to enable them to withstand strains without breaking.

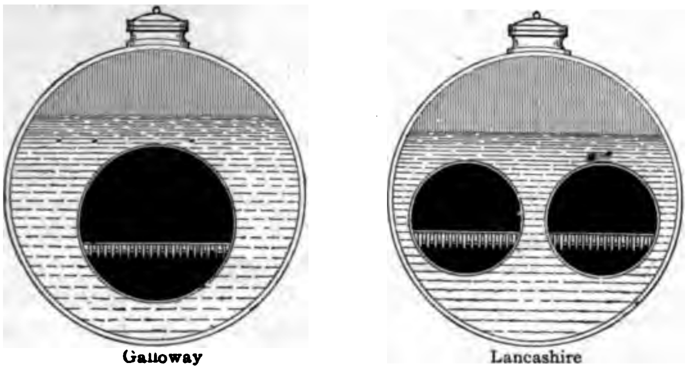


Fig. 82.—Sections of Galloway and Lancashire boilers.

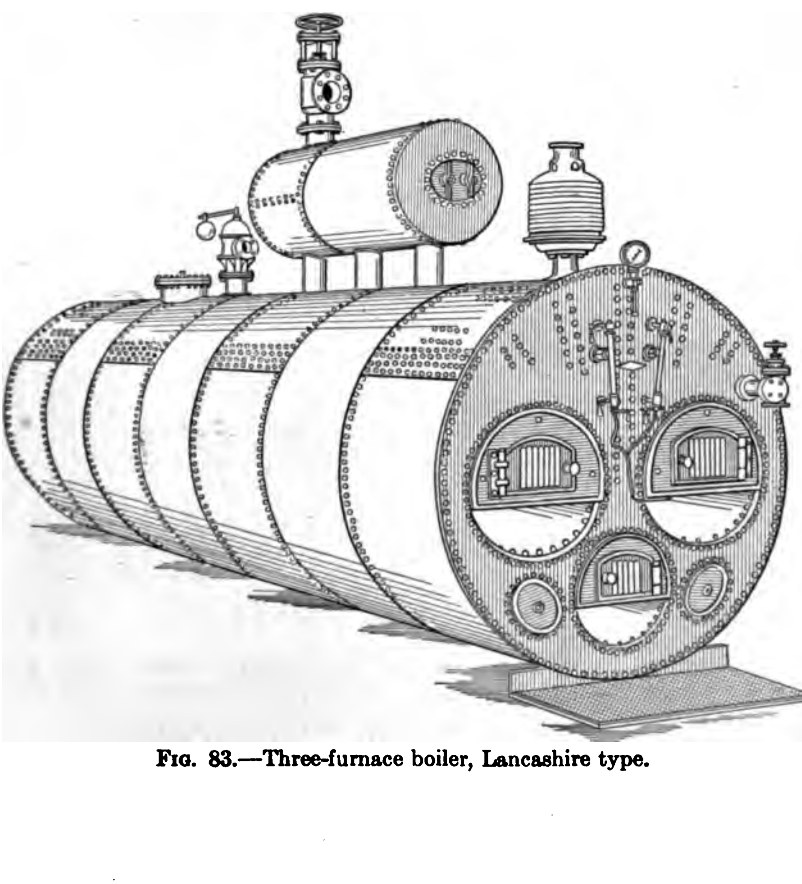


Fig. 83.—Three-furnace boiler, Lancashire type.

Five materials, copper, brass, cast iron, wrought iron and steel have been used in boiler construction but as they have all been practically forced from the field with the exception of steel, their relative advantages and disadvantages need not be presented.

It is true, however, that copper is still used to some extent for tubes for fire engines and for fireboxes in some foreign locomotives; that brass is similarly used but seldom found in practice today; that cast iron is restricted to a few types of low-pressure sectional boilers and to small parts of larger boilers; that wrought iron possesses high tensile strength and the necessary toughness, elasticity and ductility. The objection to it is its tendency to laminate or blister or both. Very little wrought iron is made today, so it is seldom found in boiler construction. Today steel is practically the only boiler material in commercial use.

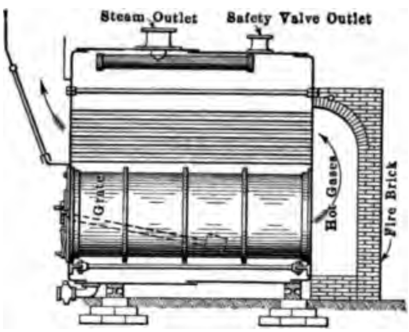


FIG. 84.—Dry-back marine-type boiler.

Specifications for Boiler Construction.[1]—The American Society of Mechanical Engineers published in 1915 a special report on Standard Specifications for the Construction of Steam Boilers and Other Pressure Vessels and for Care of Same in Service. A brief summary of the points of direct interest in connection with these notes is presented.

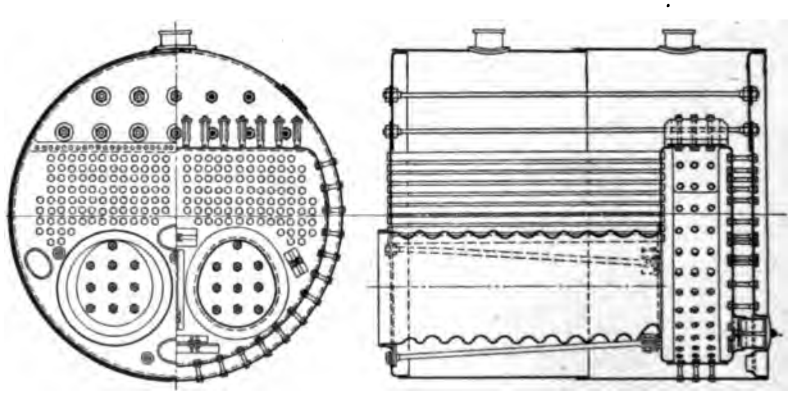


FIG. 85.—Scotch marine boiler.

Maximum Pressure.—The pressure allowed on a boiler constructed wholly of cast iron shall not exceed 15 lb. per square inch.

[1] For detailed report, see *Transactions* A.S.M.E., vol. 37, 1916.

The pressure allowed on a boiler, the tubes of which are secured to cast- or malleable-iron headers, or which have cast-iron mud drums, shall not exceed 125 lb. per square inch.

Safety Valves.—Each boiler shall have not less than two safety valves, one set for the allowed pressure and the other 3 lb. higher. Where more than two valves are used on the same boiler (as in cases of operation at 300 per cent. of rating), the additional valve or valves should be set to blow at 4 or 5 lb. higher than the first valve or valves which start to blow at the maximum working pressure allowed.

Fusible Plugs.—Each boiler shall have one or more fusible plugs.

Steam Gage.—Each boiler shall have a steam gage connected to the steam space of the boiler by a syphon, or equivalent device, sufficiently large to fill the gage tube with water, and in such manner that the steam gage cannot be shut off from the boiler except by a cock with T or lever handle, which shall be placed on the pipe near the steam gage. The connections to the steam gage shall be of brass.

The dial of the steam gage shall be graduated to not less than one and one-half times the maximum pressure allowed on the boiler.

Each boiler shall be provided with a $\frac{1}{4}$-in. pipe size connection for attaching inspector's test gage when boiler is in service, so that the accuracy of the boiler steam gage can be ascertained.

Water Glass and Gage-cocks.—Each boiler shall have at least one water glass, the lowest visible part of which shall be above the fusible plug and lowest safe water line. Shut-off valves of the outside screw and yoke type are advised in both top and bottom connections to boiler to permit of blowing through either independently.

Each boiler shall have two or more gage-cocks, located within the range of the visible length of water glass, when the maximum pressure allowed does not exceed 15 lb. per square inch, except when such boiler has two water glasses, located not less than 3 ft. apart, on the same horizontal line.

Each boiler shall have three or more gage-cocks, located within the range of a visible length of water glass, when the maximum pressure allowed exceeds 15 lb. per square inch, except when such boiler has two water glasses, located not less than 3 ft. apart on the same horizontal line.

Feed Pipe.—Each boiler shall have a feed pipe fitted with a check valve, and also a stop valve or stop-cock between the check valve and the boiler, the feed water to discharge below the lowest safe water line. Means must be provided for feeding a boiler with water against the maximum pressure allowed on the boiler.

Stop Valve.—Each steam outlet from a boiler (except safety-valve connections) shall be fitted with a stop valve.

When boilers of 50 hp. or over are set in battery, each boiler shall have two stop valves, or a stop valve and stop-cock, on the feed pipe, one on each side of the check valve.

Horsepower Rating.—A boiler having 1 sq. ft. of grate surface shall be rated at 3 hp. when the safety valve is set to blow at over 15 lb. pressure per square inch.

A boiler having 2 sq. ft. of grate surface shall be rated at 3 hp. when the safety valve is set to blow at 15 lb. pressure per square inch or less.

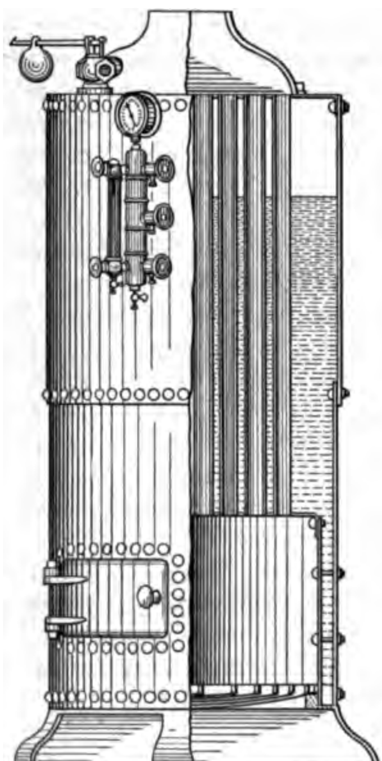


FIG. 86.—Small vertical boiler.

Power Ratings for Classification.—The horsepower of a boiler shall be ascertained upon the basis of 3 hp. for each square foot of grate surface, if the boiler is used for heating purposes exclusively. The engine power shall be reckoned upon a basis of a mean effective pressure of 40 lb. per square inch of piston for a simple engine; 50 lb. for a condensing engine; and 36 lb. for a compound condensing engine reckoned upon the area of low-pressure piston. The power rating of steam turbines shall be based on the builders' brake horsepower name-plate rating when such information is available; or when not available and steam turbine is direct-connected to electric generating apparatus shall be taken as the kilowatt rating of generator times 1.34. In case other suitable means of determining rating is lacking the chief of the State boiler department may cause such investigation as is necessary to determine the normal capacity of turbine to be made, and his decision as to rating to be allowed shall be final.

Boiler Settings.—Internally fired boilers are ready to use as soon as located and properly supported (except Cornish and Lancashire and dry-back marine).

Material used for setting boilers is brick. Parts exposed to the fierce action of heat should be of firebrick, the rest of common brick. Bridge walls and thin parts at front of fireboxes will be of firebrick.

The standard mixture for use in boiler settings is usually $\frac{1}{3}$ lime and $\frac{2}{3}$ cement. Firebrick are usually set in fire clay.

If the brick setting is to support the boiler and its contents, it must be at least $1\frac{1}{2}$ bricks (13 in.) thick, and will usually be 17 in. or 21 in. for outside walls. The 21-in. wall is usually made with a 2-in. or 4-in. air space between two walls.

Between boilers in a battery hollow walls are of no value. Such walls are usually solid and 13 in. thick.

If the boiler is supported upon iron framework, the brick shell is used to retain the heat and gases only. Under these conditions the 13-in. solid or 21-in. wall with air space is used.

Rear walls if of brick are usually solid but rarely over 12 in. in thickness.

With water-tube boilers the rear brick wall is usually displaced by steel doors.

To avoid the infiltration of air through cracks in the brick setting an air-tight steel casing has, in many instances, been adopted. The brick walls are made thinner, and are backed up by 2 in. of magnesia block and $\frac{3}{8}$ in. of asbestos board. Outside of this a thin steel casing $\frac{3}{32}$ in. to $\frac{1}{8}$ in. provided with small angles and asbestos packing is erected. The steel casing is supported by the buck-stays. As this casing is held together by small bolts any portion may be removed for repairs to the brickwork or magnesia insulation. Air infiltration is prevented and radiation and heat leakage is reduced to a minimum.

FIG. 87.—Manning boilers.

The infiltration of air has been greatly reduced by the painting of the boiler setting with a special enamel paint. This paint must not get tacky with heat or crack when cold.

Ample freedom for expansion must be allowed in setting boilers. The foundation for a boiler setting may be light if the soil is firm, otherwise a substantial foundation must be put in. Such foundations are usually of concrete although stone or brick are often used in the smaller plants.

Buck-stays and Tie-rods.—In brick boiler settings it is necessary to confine the brickwork to prevent cracking and bulging due to the heat-

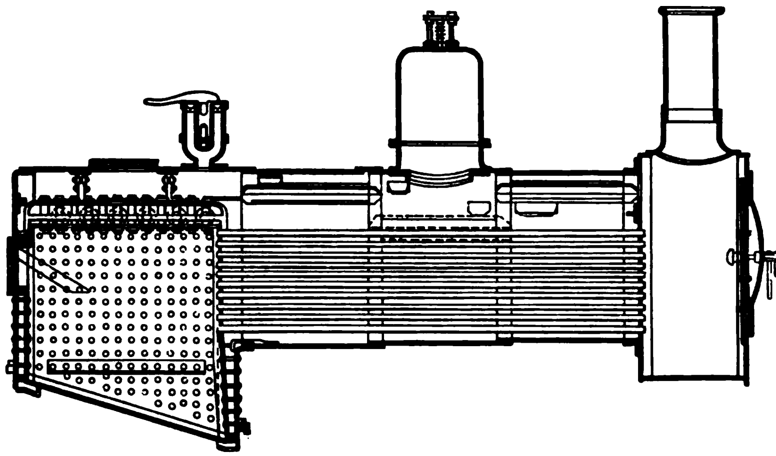


FIG. 88.—Locomotive-type boiler.

ing of the setting. For this purpose it is usual for the boiler manufacturer to supply a set of cast-iron girders which are placed vertically on the outside of the setting and tied together with steel tie-rods above

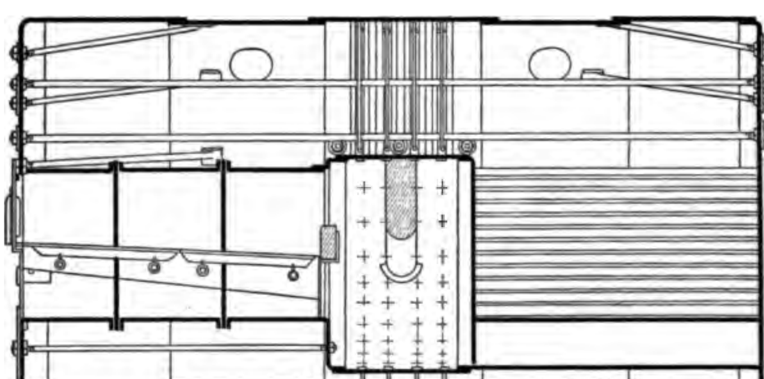


FIG. 89.—Fairbairn-type internally-fired multitubular boiler.

the boiler and below the ashpit floor. The cast-iron girders or buck-stays are usually spaced about 5 ft. apart and keep the boiler setting in shape. Steel has been used instead of cast iron for these buck-stays, especially on the larger and higher boiler settings in which case two 6-in.

channels placed back to back have been used with good success. The tie-rods are usually of 7/8 in. steel, but should be figured to take care of any thrust which the design of the boiler setting may throw on them.

Hanging or Supporting Boilers.—The ordinary return tubular boiler is supported usually on the side walls of the brick setting by means of lugs riveted to the boiler shell, slightly above the center line. These lugs are supported on round iron rollers to allow for a slight expansion

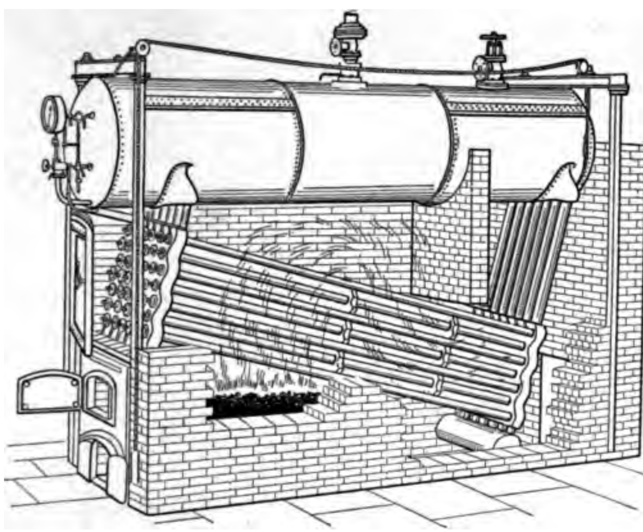


FIG. 90.—Babcock & Wilcox-type boiler.

movement. It is also common to hang boilers of this type, using the buck-stays as columns with a supporting beam across their upper ends. This makes a much more flexible and secure support and relieves the brickwork of many severe strains.

Practically all of the water-tube types of boilers are hung from an overhead structure supported on the buck-stays. The weights are so large that the brickwork would have to be inordinately heavy to support them. Internally fired boilers of the Manning or dry-back marine types are supported directly on the foundations.

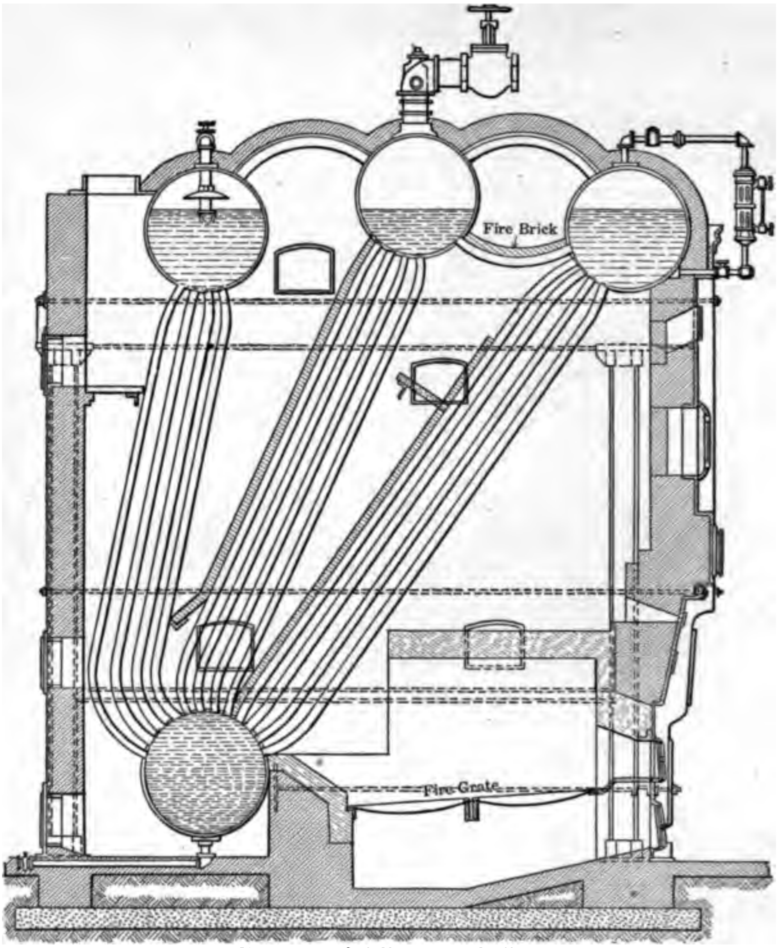


FIG. 91.—Stirling-type boiler.

Cost of Fire-tube Boilers.—

Hp.	Average size	Cost f.o.b. factory	Cost per hp., f.o.b. factory	Cost of setting	Cost of boiler set	Cost set per hp.
50	54" × 14'	$485	$9.65	$240	$725	$14.50
60	54" × 16'	565	9.42	260	825	13.75
70	60" × 14'	620	8.85	295	915	13.10
80	60" × 16'	700	8.75	300	1,000	12.50
100	66" × 16'	850	8.50	310	1,160	11.60
125	72" × 16'	1,000	8.00	385	1,385	11.10
150	78" × 18'	1,170	7.80	450	1,620	10.80
175	....	1,310	7.50	500	1,820	10.30
200	....	1,400	7.00	540	1,940	9.70

These figures may be reduced to the following approximate formulæ for the cost of horizontal fire-tube boilers.

Cost f.o.b. factory (\$) = $180 + 6.4 \times$ hp.

Cost of setting (\$) = $140 + 2 \times$ hp.

Other cost figures reported give results considerably below those of the formula above, the average of three such formulæ being

f.o.b. cost (\$) = $100 + 4.5 \times$ hp.

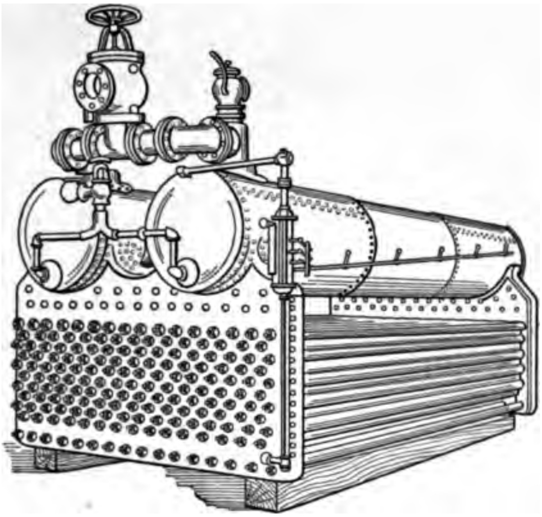


FIG. 92.—Heine-type boiler.

Cost of Water-tube Boilers.—The following table gives the average prices for several different makes of water-tube boilers f.o.b. factory.

Hp.	Total cost	Cost per hp.
100	\$1,360	\$13.60
125	1,540	12.30
150	1,730	11.50
175	1,950	11.15
200	2,200	11.00
250	2,600	10.40
300	3,100	10.30
400	4,100	10.20
500	5,000	10.00

From these figures the following formula is derived:

$$\text{Cost f.o.b. factory (\$)} = 425 + 9 \times \text{hp.}$$

For want of better average figures on price of settings for water-tube boilers, the formula for settings for fire-tube boilers may be used, viz.:

$$\text{Cost of setting (\$)} = 140 + 2 \times \text{hp.}$$

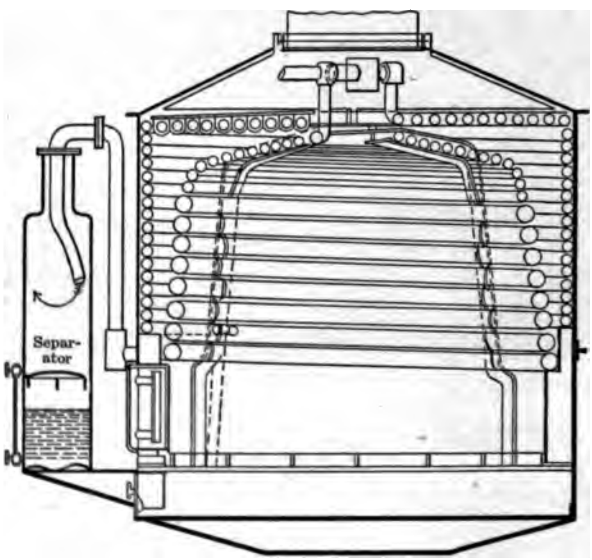


FIG. 93.—Herreshoff boiler.

Another estimate for the cost of the brickwork for boiler settings is:

$3.50–$3.00 per hp. up to 100 hp.
2.50– 2.00 per hp. up to 200 hp.
2.00– 1.00 per hp. above 200 hp.
($1.00 at 500 hp.)

Potter's formulæ (*Power*, Dec. 30, 1913)—slightly modified—for the cost in dollars of the usual sizes of water-tube boilers are:

For vertical boilers $= 900 + 6.3 \times$ hp.

For horizontal boilers $= 150 + 8.2 \times$ hp.

The following table gives the prices of high-grade water-tube boilers, f.o.b. factory in 1916.

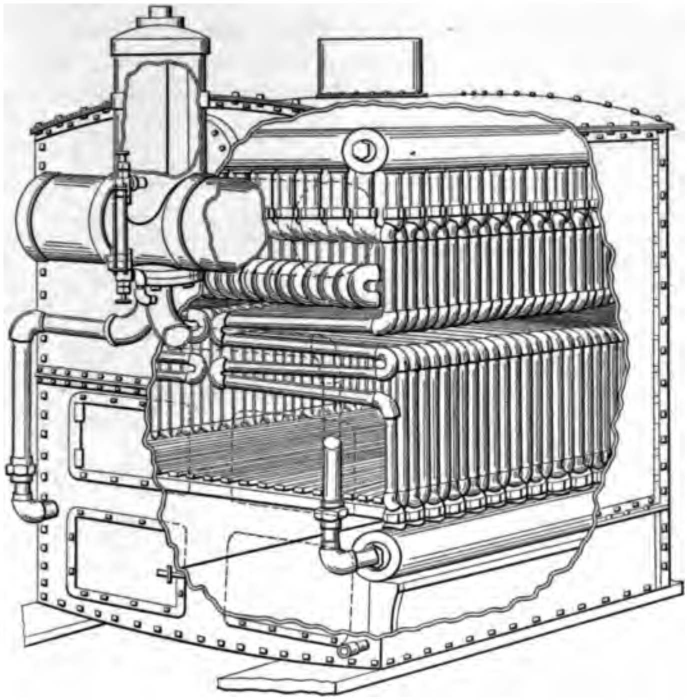


FIG. 94.—Almy water-tube boiler.

HORIZONTAL WATER-TUBE BOILERS WITH CAST-IRON HEADERS, 160 LB. PRESSURE

	Single boiler		Two in battery	
Hp. each boiler	Total cost	Cost per hp.	Total cost	Cost per hp.
200	$3,000	$15.00	$5,900	$14.75
300	4,000	13.35	7,750	12.90
400	4,800	12.00	9,350	11.70
500	6,000	12.00	11,350	11.35
600	7,200	12.00		

HORIZONTAL WATER-TUBE BOILERS WITH WROUGHT-STEEL HEADERS, 200 LB. PRESSURE

	Single boiler		Two in battery	
Hp. each boiler	Total cost	Cost per hp.	Total cost	Cost per hp.
200	$3,900	$19.50	$7,600	$19.00
300	5,000	16.70	9,600	16.00
400	5,800	14.50	11,500	14.40
500	7,050	14.10	14,000	14.00
600	8,750	14.60		

Average Cost of Boilers.—The average cost of boilers including setting, as reported by one consulting engineer, on the basis of boiler capacity required for the indicated horsepower rating of the plant is:

COST OF BOILERS PER ENGINE HORSEPOWER
(Including Setting)

Simple non-condensing:									
Engine horsepower	10	12	14	15	20	30	40	50	75
Cost per horsepower	$56.00	$51.00	$46.00	$43.50	$32.00	$26.00	$24.00	$20.50	$17.30
Simple condensing:									
Engine horsepower	10	12	14	15	20	30			
Cost per horsepower	$35.50	$33.10	$29.60	$28.50	$25.10	$20.50			
Engine horsepower	40	50	75	100					
Cost per horsepower	$17.80	$15.80	$14.80	$14.20					
Compound condensing:									
Engine horsepower	100	200	300	400	500	600	700	800	
Cost per horsepower	$8.00	$7.60	$7.40	$7.30	$7.25	$7.20	$7.15	$7.05	
Engine horsepower	900	1,000	1,500	2,000					
Cost per horsepower	$6.90	$6.80	$6.60	$6.40					

Furnace Design.—There is no other apparatus in a power plant upon which so much depends as the boiler furnace. This is usually the place to look for increased economy. Considering the design of the furnace, there are three essential features:

First.—A grate must be provided upon which to burn the coal.

Second.—Means must be provided for the admission of a proper amount of air to facilitate combustion.

Third.—A combustion chamber must be installed, of the proper shape and capacity, for the gases which are to be burned.

Flat grates are used when coal is fired by hand. They may be classified as shaking and dumping grates. The shaking grate is best adapted for burning coal which has a limited amount of ash, and which does not clinker badly. The dumping grate is adapted, as well as any, to practically all kinds of fuel, though generally it is more expensive than the shaking grate.

When forced draft is installed, the air is generally admitted in the front of the ashpit which forms a chamber large enough to allow the air to come to rest, the pressure in this chamber forcing the air through the fire more or less uniformly. When the induced draft system is used, the ashpits are left open and the air is drawn in under the grate, or through the fire-door, or at any other place where there may be a leak.

When forced draft is produced by a steam jet—which is frequently used in very small installations—it is common practice to place this steam jet in the side wall in the ashpit. These steam jets are commonly believed by the operating engineers to be especially advantageous on account of the prevention of clinker in the ash. This is a very poor

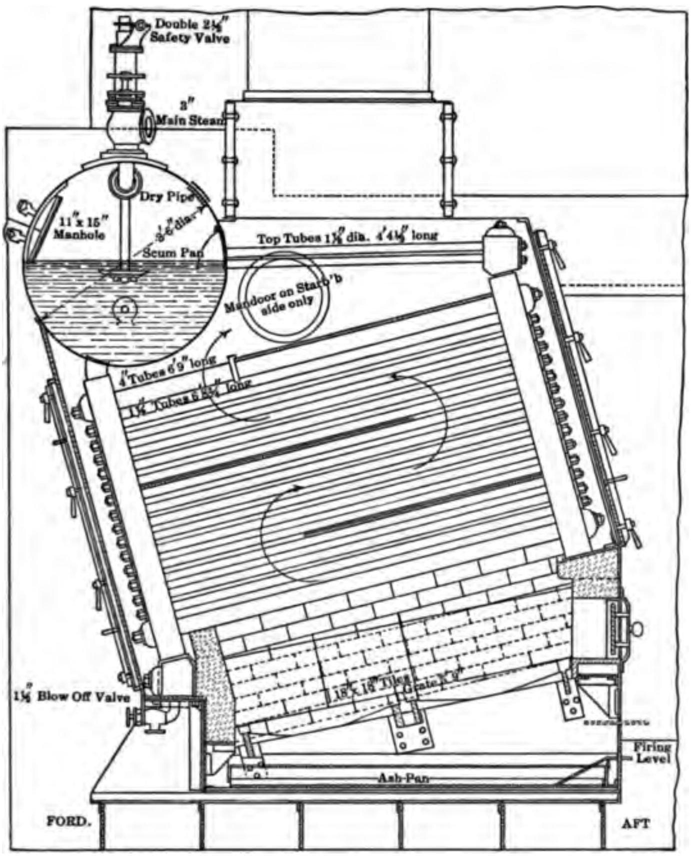


FIG. 95.—Babcock & Wilcox marine boiler.

means for preventing clinker, however, as clinker formation is reduced because the steam cools the fire below the clinkering temperature. Recently a combination turbine-driven disk-fan has been used, which is very well adapted to these small installations on account of the fact that a much less percentage of steam is passed into the grates along with the air. The warm air is a great help to the combustion of the coal, but few attempts have been made to preheat the air before introducing it under the grate.

One system which has been used to a small extent, is that of forcing the air through passages in the bridge-wall and side walls, but these chambers gradually become clogged, and the maintenance expense is usually too high to make the advantage derived from the warm air worth while. For internally fired boilers the Howden and Ellis-Eaves systems are much used in marine practice and also in stationary practice abroad. In Howden's system the air, driven by a fan, is forced through tubes in the uptake. Here it is heated and then led into the ashpit and also above the grate. The Ellis-Eaves system includes the induced draft principle and is usually applied to boilers with larger tubes than the Howden system.

The most important consideration in the design of the furnace is in the combustion chamber. The more volatile matter the coal contains, the more difficulty this problem presents. In burning the fine grades of anthracite, it is possible to have the heating surfaces of the boiler relatively close to the grate, as practically no hydrocarbons are distilled from this coal. With horizontal multitubular boilers, the distance from the top of the grate to the underside of the boiler is not over 2 to 3 ft., and in many cases, under these conditions, this coal is burned with fairly good results. When soft coal, however, is burned in the same furnace, it invariably produces a large amount of smoke, and the furnace efficiency is very low. When the hydrocarbon content of the coal is distilled off, before it can be sufficiently mixed with the air to burn, it comes into contact with the surface of the boiler, and is chilled to a point below the combustion temperature. To prevent this, an arch or arches may be provided under the shell of the boiler, or the furnace may be built in the form of the well-known Dutch oven. A very good setting is produced by springing several arches across the grate, between the side walls of the boiler, leaving an area between these arches, through which the gases may pass, about equal to the area blocked by the arches.

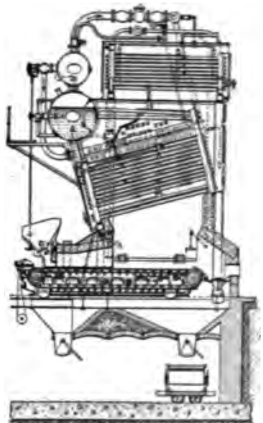


FIG. 96.—Boiler, economiser, superheater and stoker at the Eberswalde Station, Markische E. W.

The height at which the boiler shell should be set above the grate depends upon the amount of volatile constituent in the coal. With coal containing from 15 to 20 per cent. volatile matter, this distance should be not less than 4 to 5 ft.; and if the coal contains a larger amount of volatile matter, this distance should be increased as far as possible.

The Dutch oven, of course, will produce a better result than the construction just described, but it is more expensive to install, and it must be built out in front of the boiler, taking up a considerable amount of space. The maintenance is also probably in excess of that of the arches.

When water-tube boilers are installed, the design of the furnace takes a somewhat different form. In the first place it is always much wider than for fire-tube boilers; and, again, constructive considerations make it easier to obtain the required volume for the combustion chamber when burning highly volatile coal. In burning fine grades of anthracite, in many cases a plain, flat grate only is provided, and this gives fairly good results.

The Webster furnace which has been used in some very large installations of late years, is designed, primarily, for burning fine anthracite, and has given excellent satisfaction. This furnace consists of four firebrick arches sprung across the grate between the side walls of the boiler, their object being to prevent the cooling of the fire when the doors are opened for firing. This is a very important matter, especially when induced draft is used, as in this case the pressure of the atmosphere is anywhere from $\frac{5}{10}$ to 1 in. of water above that over the fire; and when the doors are open an immense amount of cold air will rush in, chilling the tubes and cooling the fire. When forced draft is in use, the difference in pressure between the air outside and that over the fire is only sufficient to carry away the flue gases, and is seldom over $\frac{1}{10}$ or $\frac{2}{10}$ in. of water, as the air is forced through the fire by the draft.

For burning bituminous coal under the water-tube boiler, the number of different types of furnaces used is almost infinite. With the boilers of the Babcock and Wilcox type, the Murphy furnace—which is essentially a Dutch oven—is frequently applied, and generally produces good results. These boilers are, however, most always used with some form of mechanical stoker, and will be considered later.

The plain Dutch oven is sometimes used, but only on coals of relatively high volatile content. One of the most effective ways of increasing the volume of the combustion chamber, and keeping the volatile gases from contact with the water surfaces of the Babcock and Wilcox type of boiler, is to provide a baffle covering the lower portion of the first and second passes, thus circulating the products of combustion first over the bridge-wall, thence up through the third pass, down the second, and up the first pass to the uptake. This makes possible a firebrick roof over the whole grate, extending some 3 or 4 ft. back of the bridge-wall, and has all the effect of a Dutch oven without the disadvantages of the latter. The scheme does not require any extension front built out from the boiler.

A very important consideration, irrespective of the type of furnace put in, is the height of the boiler tubes above the floor or, what amounts

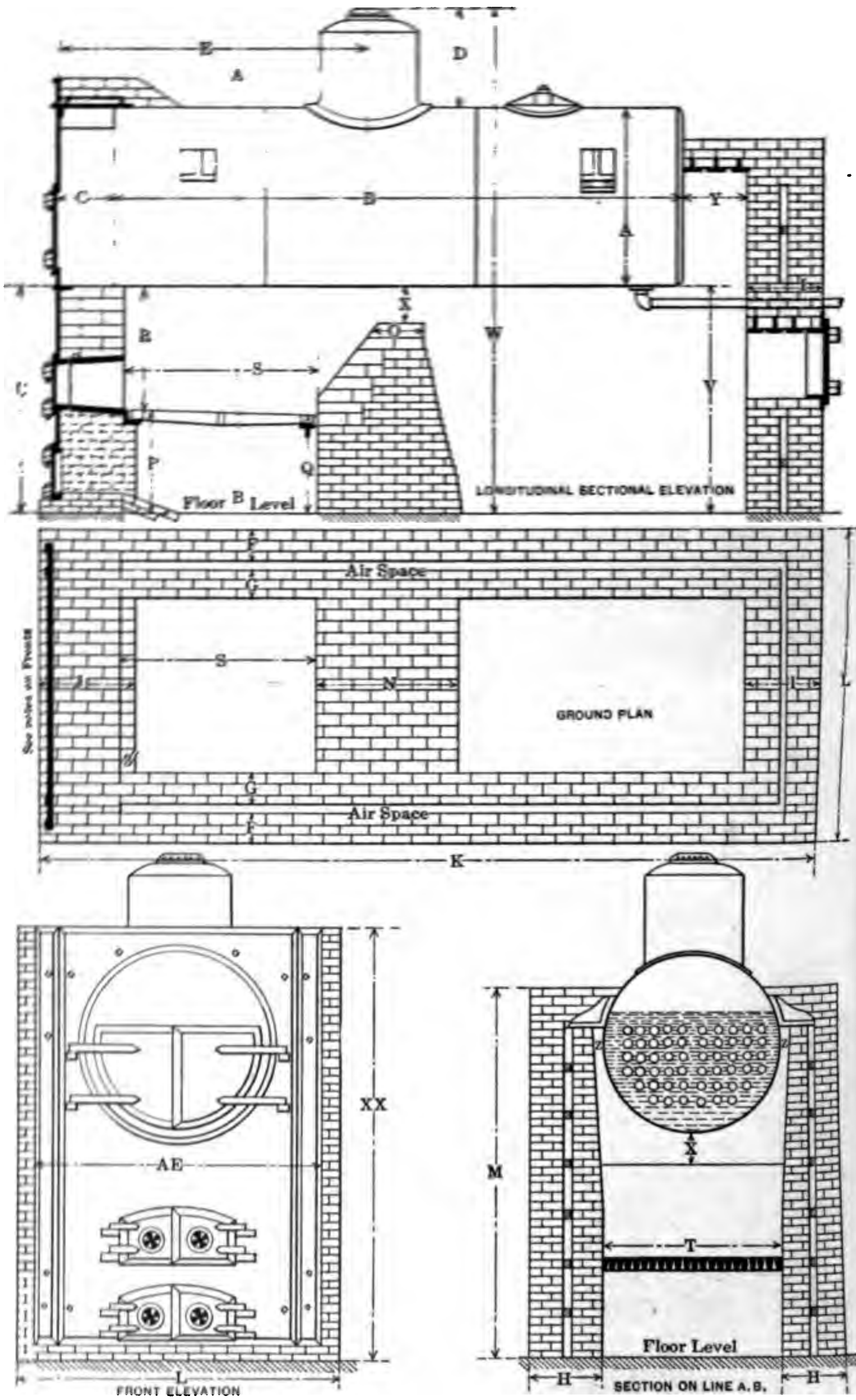


FIG. 97.—Setting for flush-front boilers.

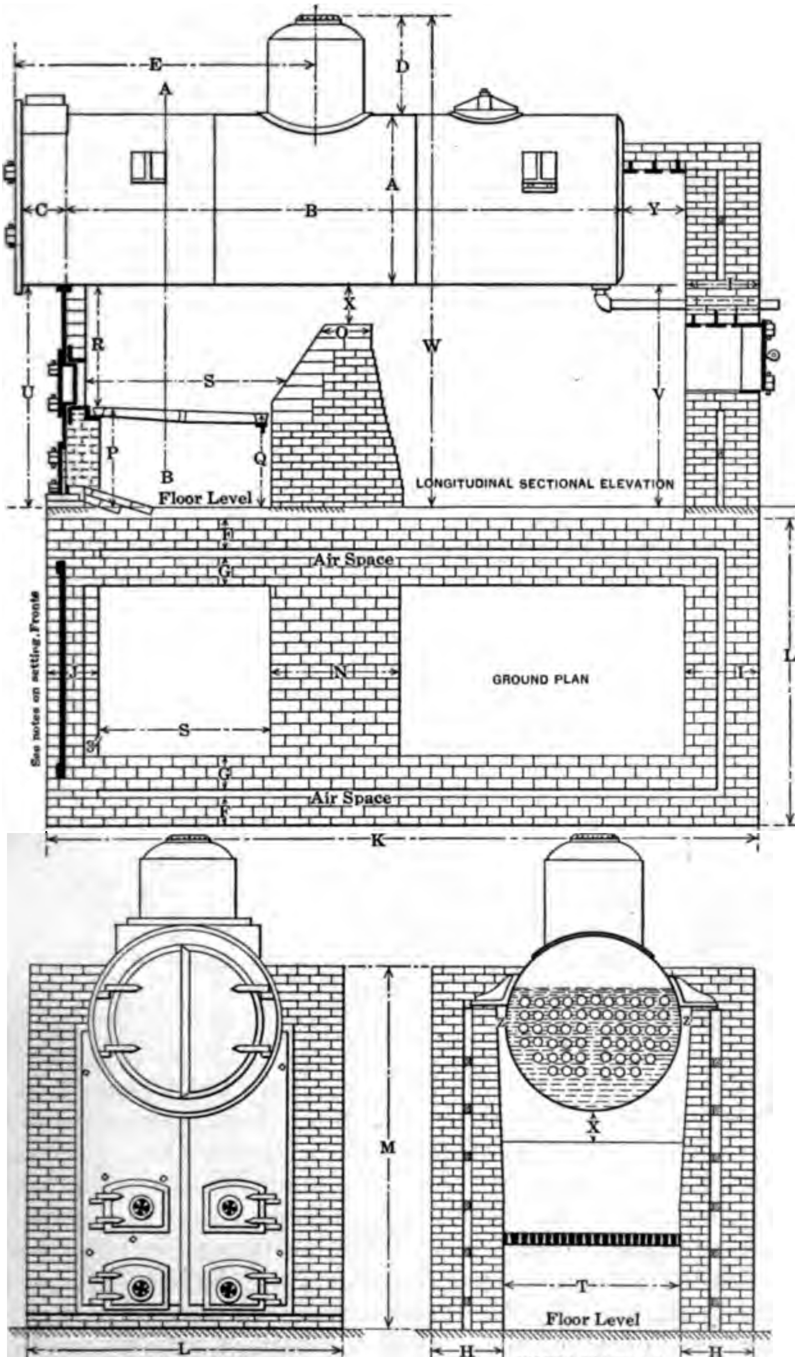


Fig. 98.—Setting for arch-front boilers.

to the same thing, above the grate. Formerly it was considered customary to install the B. & W. boiler with the bottom of its front header about 7 ft. 6 in. above the floor line. This distance has been gradually increased for burning the high-volatile bituminous coals, to 9 ft.; and now, in some of the most recent installations, has been made 10 to 12 ft. above the floor line. Even this figure might well be increased for burning western coals which contain 30 per cent. or more volatile matter.

Another type of furnace, although not very widely used, for which a great deal is claimed, is the Hawley down draft furnace. This con-

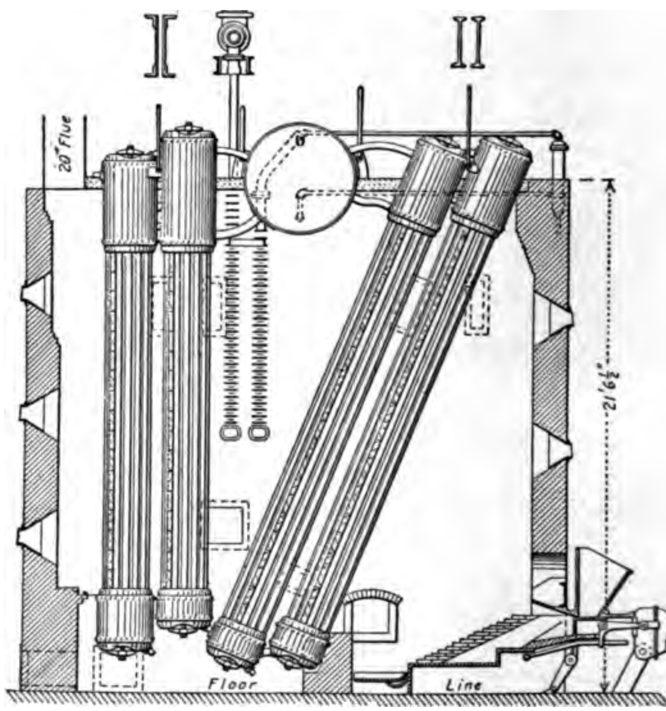


Fig. 99.—Bigelow-Hornsby boiler.

sists, essentially, of an ordinary flat grate above which, some 2 ft. or more, is a secondary grate composed of tubes through which the boiler water circulates. The coal is fed onto this secondary grate. The gases are distilled from it here, breaking the coal up and allowing it to fall through the bars onto the main grate. The gases from the fresh coal, as it is coked on the secondary grate, pass down through the incandescent coal which lies underneath, and thence into the combustion chamber. Here the gases, which by this time are more or less thoroughly mixed with air, are burned by the heat of the burning coke on the main grate below.

In considering the amount of coal which can be burned per square foot f grate surface, we find variations under different conditions of from 2 to 125 lb. per hour. In central-station practice, burning the finer

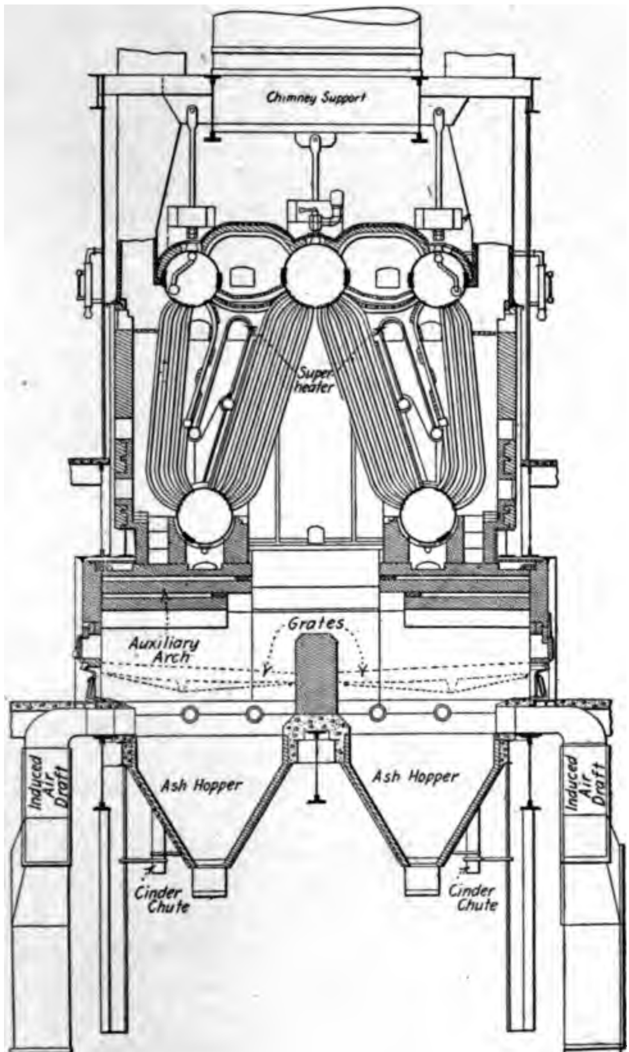


FIG. 100.—Large Stirling boiler as installed at Hauto, Pa.

grades of anthracite where forced draft is almost universally employed, it is customary to burn, under normal conditions, in the neighborhood of from 25 to 30 lb. of coal per square foot of grate. On the peak loads by increasing the draft on the fire, this figure may be, and very often is, increased to 50 lb. per square foot. The economy when this latter amount of coal is burned, is somewhat diminished, due largely to the fact that the draft necessary to burn this amount of coal is sufficient to cause a certain percentage of it to pass into the back combustion chambers of the boilers, and to be deposited in the flues unburned, causing considerable loss.

Soft coal is very seldom burned on flat grates in any of the reasonably large central power stations, some form of mechanical stoker being employed. When soft coal is burned on flat grates, however, it is usually customary to burn not over 20 lb. per square foot, as it is very difficult with hand-firing, to produce an even distribution of air, and in consequence, if sufficient time is not given the air to mix with the volatile gases, a large part of these will pass off unburned.

Furnace Losses.—The most serious furnace losses are:

(*a*) Heat necessary to create draft.
(*b*) Heating air used for combustion.
(*c*) Heating ash.
(*d*) Radiation.
(*e*) Incomplete combustion, smoke, etc.
(*f*) Imperfect transfer of heat.

Mechanical Stokers.—*The Stoker Problem.*—In considering whether or not mechanical stokers should be installed in a proposed plant many technical factors and local conditions must be taken into account.

The kind of coal to be burned and its cost are important factors. In general it may be stated that unless the firemen are expert and exceptionally well handled, coal will be burned more economically by using a reasonably well-designed mechanical stoker than by the hand-fired method. The quality of the labor at hand if not of a good character gives added weight to the stoker as it does not require such intelligent firemen to handle the stokers properly as it does to fire the coal by hand with equal efficiency. The labor problem is also important in the larger plants on account of the possibility of strikes. In small plants it is always a question whether the saving in labor and coal will warrant the investment for stokers. In localities where the finer grades of anthracite, such as the buckwheat sizes, are available at a low price it will generally be more economical especially in the smaller plants to fire the coal by hand. In the larger plants the price of the coal must be weighed carefully against the other conditions previously noted and the decision must be made on the merits of the individual case.

There is one other point which perhaps gives some advantage to hard coal and that is when large storage capacity is required. Hard coal may be stored in practically unlimited amounts, while it is well known that great difficulty due to spontaneous combustion is experienced in storing arge quantities of bituminous coal especially when it runs high in volatile content. A number of plants for the storing of bituminous coal under water to prevent loss by spontaneous combustion have been installed.

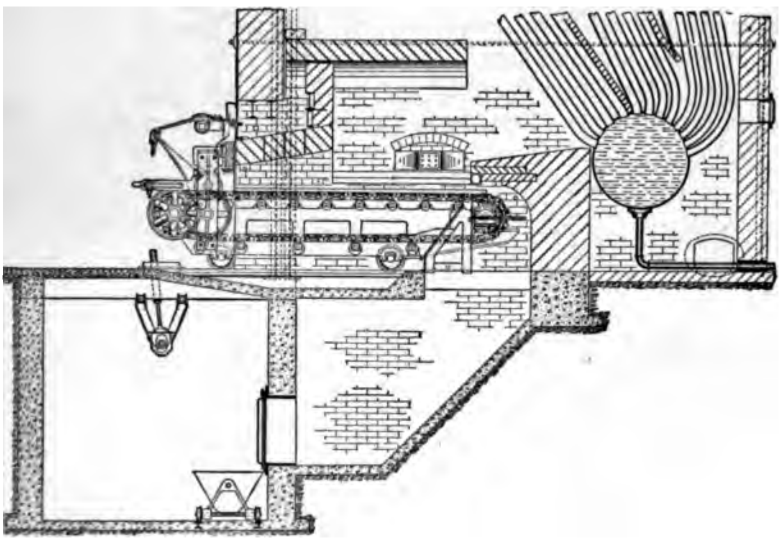


FIG. 101.—Green chain-grate stoker under Stirling boiler.

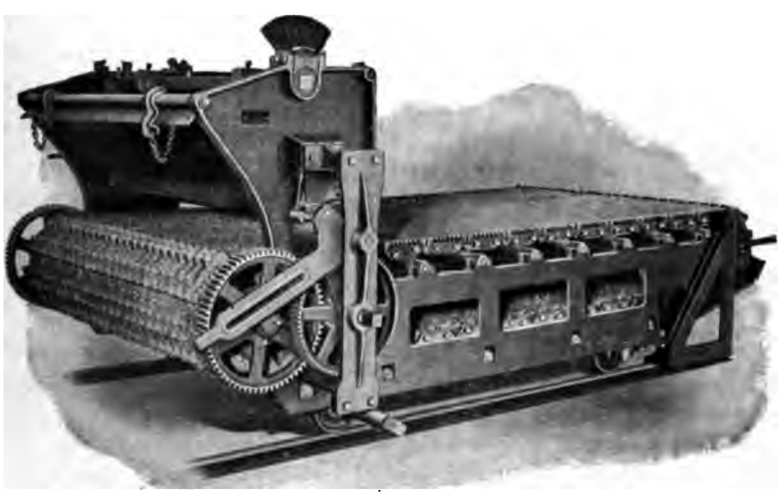


FIG. 102.—Green chain-grate stoker.

Types of Stokers.—There are three general types of mechanical stokers, although there are several which cannot be conveniently placed in any one of these classes.

The first is the chain-grate stoker consisting of an endless chain placed in the furnace of the boiler with its top side revolving slowly from the front of the furnace toward the back. The coal is fed onto this moving grate in front and is burned as it passes toward the bridge-wall, where, when the grate is moving at its proper speed, the coal will have been completely burned to ash which will drop down into the ashpit below. Some representative stokers of the chain-grate type are the Babcock and Wilcox, the Green, the Playford and the American.

The second type of stoker is the inclined overfeed type, of which the Roney, the Ross, the Murphy and the Wilkinson are examples. These stokers consist of movable bars forming an inclined grate with a

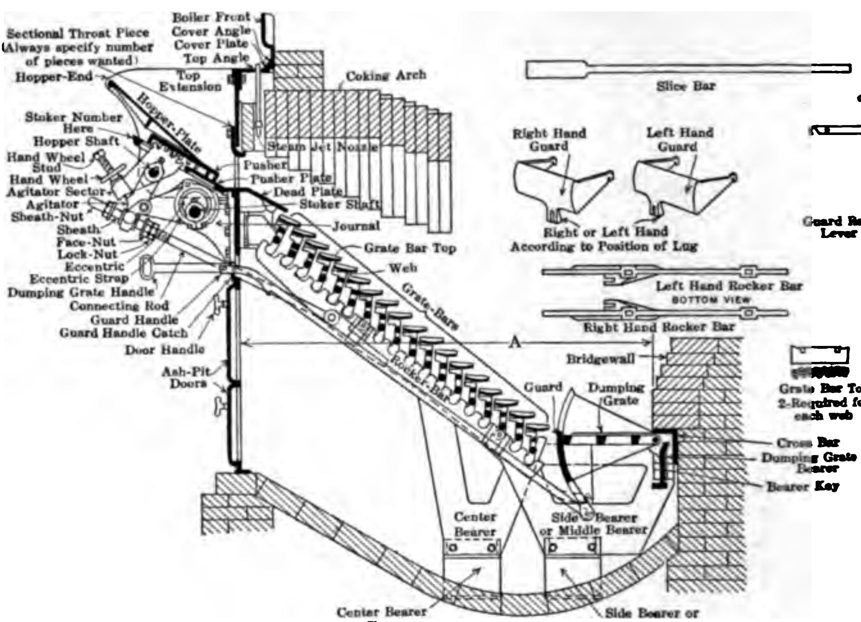


FIG. 103.—Section of Roney stoker.

mechanism for moving these bars in such a manner as to cause the coal which is fed in at the top to be pushed gradually down the grates until it reaches the dumping grate at the bottom, at which point it should be completely burned.

The third type is the underfeed of which the Taylor, the Jones and the American are typical examples. In these stokers the coal is fed onto the grate or up under the grate in such a manner that the fresh coal is always close to the grate while the coal which is being burned is at the top. The air is also introduced at the bottom and while passing up through the bed of coal is heated and thoroughly mixed with the volatile gases distilled from the coal and when passing through the incandescent layer at the top reaches its temperature of combustion, so that generally no arches are necessary.

There are several other stokers which are used to a greater or lesser extent and work on somewhat different principles, perhaps the most notable being the so-called "finger" stoker which has several oscillating paddles which pick up the coal and throw it into the furnace. There is another stoker somewhat of this order in which the coal is mechanically shovelled into the furnace. These stokers, however, have not reached any extensive application as yet and in general are only applicable to particular conditions.

Chain Grates.—The one great advantage of the chain-grate stoker over most of the others is that it is suitable for burning with natural draft

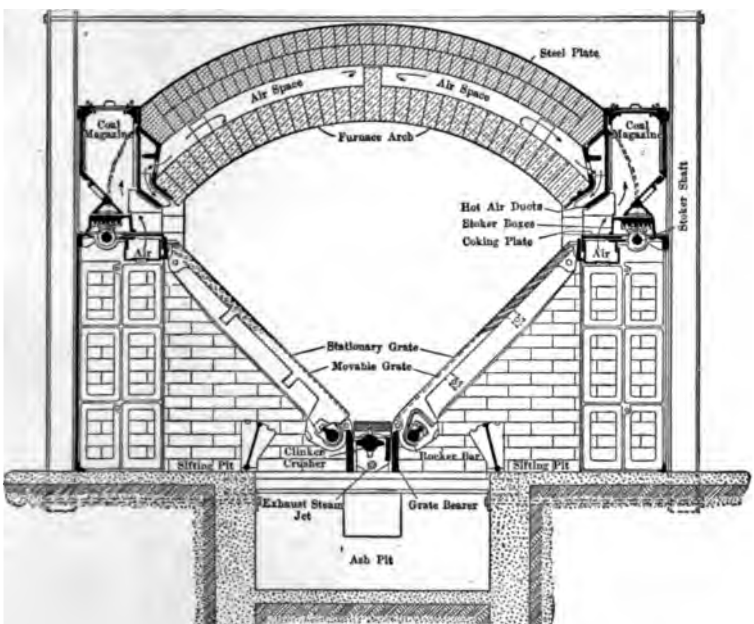


FIG. 104.—Murphy stoker.

bituminous coals of very high volatile content. In plants where the load is reasonably steady and where sudden peaks are not thrown upon the boilers this stoker makes a first-class installation.

There are two serious troubles with this type of stoker which always have to be contended with. The first of these is the leakage around the back of the stoker which, in most installations, is very large and cools the combustion chamber beyond the point at which volatile gases are ignited. This has been obviated to a large extent by certain of the manufacturers by placing a long flat arch from the bridge-wall toward

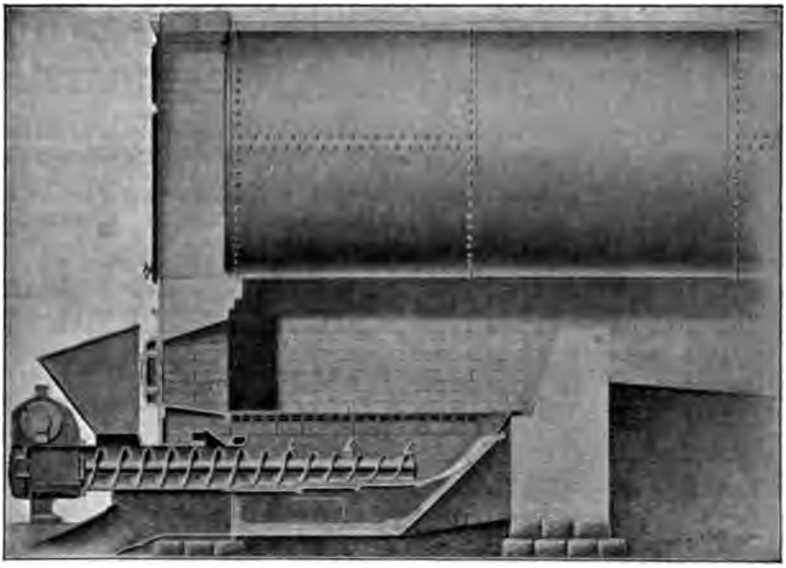


Fig. 105.—American underfeed stoker.

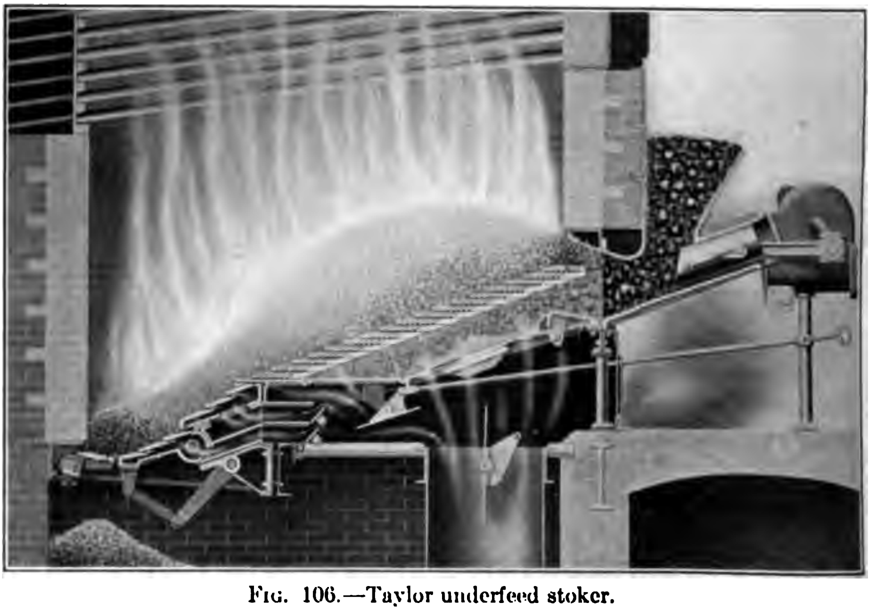


Fig. 106.—Taylor underfeed stoker.

the front so that the air which pours up from the back of the chain grate will be heated to a sufficient temperature so that it will not cool off the combustion chamber. The other trouble which must be looked after carefully is the very large percentage of combustible in the ash which always occurs with a chain-grate stoker in spite of the fact that the ash-pit only covers, in most cases, less than one-third of the area under the grate.

The method of feeding the coal on to the moving grate varies in the different makes. In the Playford and the Babcock and Wilcox stokers no feeding device is used, the coal coming from the hopper directly onto the grate. In the Green stoker the coal from the hopper is coked on a series of movable bars which feed it down onto the chain grate. The American stoker has no device for feeding the coal to the chain grate but at the bridge-wall end a separate dumping grate is used instead of allowing the ashes to merely fall over the end of the grate. The coking arches vary but very little in all of these makes, these variations being due primarily to the different volatile contents of the coals to be burned.

Combustion.—The student will find available an unlimited amount of good material relating to combustion. One of the most satisfactory presentations of this subject is that of H. deB. Parsons in his book on "Steam Boilers"[1] from which much of the following material is taken.

"When heat is applied to coal, the resulting combustion is effected as follows: first, the absorption of heat; second, the vaporization of the bituminous or hydrocarbon portion and its combustion; and third, the combustion of the solid or carbonaceous part. These actions are entirely separate and distinct, and must take place in the order as given. The hydrocarbon or bituminous portion consists of marsh gas, olefiant gas, tar, pitch, naphtha, etc.

The flame is derived from the gaseous portion, and this explains why the soft or bituminous coals burn with more flame than the anthracites.

Coal gas, taken by itself, is not inflammable, as a lighted taper placed in a jar of coal gas will be extinguished. In order to consume it oxygen must be supplied, that is, the gas must be mixed with air. When this is done the gas will be consumed instantly, provided the proper temperature be present.

When a charge of fresh coal is thrown on a fire we cannot control the amount of gas that may be generated, but we can control the supply of air. Therefore it is essential, when soft coals are to be burned, that a certain amount of air be admitted in addition to the regular supply through the grate, during the periods of evolution of the gases. This can be accomplished by permitting air to enter above the grate, or directly into the combustion chamber behind the bridge wall, or both. The quantity admitted should bear some suitable relation to the percentage of the hydrocarbons contained in the fuel. It is best in all cases to provide ample passages for the air, and then to admit the proper quantity as determined by trial and observation of the smoke produced.

In order to burn the coal economically, it has been found necessary that an

[1] "Steam Boilers," by H. deB. Parsons, 4th edition, p. 15.

excess of air should be allowed to enter the furnace. If only the theoretical quantity be supplied, a large proportion of the carbon will either not be consumed or be only half burned to carbon monoxide (CO).

On the other hand, too great an excess, as well as a deficiency of air, is a detriment to the economical working of the furnace.

Much depends upon the design, especially with soft coals, for the requisite quantity may be supplied in a manner as not to be available; that is, the particles of oxygen may not come into contact with particles of carbon. In short, the air and particles of fuel may not mix, but rush to the chimney in "stream-lines."

"The temperature at which some of the physical and chemical changes take place when a fresh charge of coal is thrown on a fire are about as follows:[1]

(*a*) Previous to putting on a charge of coal the temperature of the bed of coals is from dull red heat (700°C. or 1,292°F.) up to a bright white heat (1,400°C. or 2,552°F.) or even higher.

(*b*) The coal, when fired, is about 15°C. or 60°F. (temperature of the room). As soon as it reaches a fire-bed it begins to heat by conduction from the hot coals beneath. The hot gases, products of combustion of the coal beneath, also heat the new charge of coal.

(*c*) The heating of the coal causes the volatile matter to distil off. The amount distilled at any given temperature is unknown, but it is certain that traces of volatile combustible matters are given off as low as 110°C. (220°F.).

(*d*) At about 400°C. or 750°F. the coal reaches the temperature of ignition and burns to carbon dioxide.

(*e*) At about 600°C. or 1,100°F. most of the gases given off by coal (hydrogen, marsh gas and other volatile hydrocarbons) will ignite if oxygen be present.

(*f*) At 800°C. (1,470°F.) the carbon dioxide, as soon as formed from the coal, will give up one atom of its oxygen to burn more coal, thus: $CO_2 + C = 2CO$. The carbonic oxide will burn back to carbon dioxide if mixed with oxygen at the necessary temperature, which is between 650° and 730°C. (1,200° and 1,350°F.).

(*g*) At about 1,000°C. or 1,832°F. the H_2O formed by the burning of the hydrogen in the volatile matter in the coal begins to dissociate.

(*h*) At about 1,000°C. or 1,832°F. any carbon dioxide not previously burned to carbonic oxide begins to dissociate to carbonic oxide and oxygen.

(*i*) The various hydrocarbons which begin to be distilled at 110°C. and possibly lower, undergo many changes, dissociations and breakings up at the various temperatures they pass through. So many of these are unknown that it is useless to state the few we do know.

About 700°C. (1,300°F.) both the hydrocarbons and the carbonic oxide will unite with oxygen if the latter be present and intimately mixed with them. If they do not burn, the tendency is always to break up into simpler and more volatile compounds as the temperature rises.

The composition of the gases from combustion may be found in almost any ratio. The following volumetric analyses will afford some idea of the ratio found. The last two are given on the authority of George H. Barrus, the last one being the products from Pocahontas (semi-bituminous) coal:

[1] Steam Users' Association, Boston, *Circular* No. 9, R. S. HALE's Report on Efficiency of Combustion.

	Poor. Per cent.		Average. Per cent.	Excellent. Per cent.
Carbon dioxide (CO_2)	8.0	9.0	12.0	15.1
Oxygen (O)	4.4	11.5	7.5	4.0
Carbon monoxide (CO)	7.6	Trace	0.1	0.7
Nitrogen, vapor of water, etc., by difference	80.0	79.5	80.4	80.2
	100.0	100.0	100.0	100.0

These gas analyses can be made by the Orsat or some similar apparatus, by tapping the flue and extracting a measured volume by means of a pressure bottle, such as is used in a chemical laboratory, and a graduated burette. The sample is then forced in succession through three pipettes containing caustic potash, pyrogallic acid and caustic potash, and cuprous chloride in hydrochloric acid, which will absorb respectively the carbon dioxide, the oxygen and the carbon monoxide. The loss of volume at each operation is measured in the burette.

The refuse from a fuel is that portion which falls into the ashpit and that carried off by the draft, consisting of ashes, unburnt or partially burnt fuel and cinders.

Loss by Unburned Coal in Ashpit

Remarks—authority	Per cent. refuse	Per cent. combustible in refuse	Per cent. in total coal
E. B. Coxe (*Trans.* N. E. Cotton Mfg. Assn., 1895), using his traveling grate, on small-sized anthracite coal	10.05	18.68	2.2
	23.70	11.92	2.7
W. H. Bryan (*Trans.* A. S. M. E., vol. 16, p. 773), using soft coal	13.35	31.0	4.3
	14.31	25.0	3.6
Pennsylvania coal, bars 1⅝ in. wide, 1 in. apart	16.10	25.0	4.0
Other tests	10.30	37.2	3.8
Other tests	9.20	31.3	2.9
Other tests	18.50	29.3	5.4
Other tests with mechanical stoker	13.61	67.8	9.2
Other tests with mechanical stoker	18.70	67.2	12.6
Arkansas State Geological Survey *Report*, 1888, vol. 3, p. 73, Pittsburgh coal	8.10	26.0	2.1
Ditto Arkansas coal	10.30	30.0	3.1
Ditto Arkansas coal	40.00	83.0	33.2
Ditto Arkansas coal	14.00	51.4	7.2
Dampfkessel Revision Verein Berlin Geschafts Bericht, 1895, p. 79. Coal dust	4.8	50.0	2.4

The following is from a report of R. S. Hale, Steam Users' Association, Boston, *Circular* No. 9:

"The amount of loss by unburned coal in the ashpit depends on so many factors that it is impracticable to express it by any formula. A statement of the factors and a collection of examples must, therefore, suffice.

(*a*) The loss by unburned coal in the ashpit depends on the width of the opening in the grate bars, and increases as the width increases.

(*b*) The loss depends on the size of the coal, and increases as the size of the coal decreases.

(*c*) The loss is probably greater for a non-caking than for a caking coal.

(*d*) The loss probably increases as the amount of earthy matter in the coal increases, but not the same ratio.

(*e*)[1] The loss is less with a fan blast than with a steam blast.

(*f*)[2] The loss is greater the more the fire is disturbed. This is especially noticeable in automatic stokers with moving grate bars."

The following formula of Dulong is convenient for determining the theoretical quantity of air that is required for the combustion of any fuel whose composition is known.

Let C, H and O denote respectively the weight of carbon, hydrogen and oxygen in the fuel; and W and V the weight and volume of air required. Other ingredients may be neglected, as they have but a slight effect on the result. Then

$$W = 11.61C + 34.78\left(H - \frac{O}{8}\right), \text{ or}$$

$$W = 12C + 35\left(H - \frac{O}{8}\right), \text{ nearly; and}$$

$$V = 152.56C + 457.04\left(H - \frac{O}{8}\right), \text{ or}$$

$$V = 153C + 457\left(H - \frac{O}{8}\right), \text{ nearly.}$$

The value of W per pound is about 12 for anthracite and good bituminous coals, 6 for wood, and 11 for charcoal.

It is found impossible in practice to obtain complete combustion unless the air supplied to the furnace be in excess of that theoretically required. Experience dictates that for ordinary natural draft nearly twice the theoretical quantity of air should be admitted, or about 24 lb. per pound of coal. With mechanical drafts and with natural drafts when the mixing effects are strong and positive, the excess of air may be considerably reduced.

The volume of air supply per pound of coal, in ordinary factory practice, with natural draft is about 300 cu. ft.; and may be as low as 200 cu. ft. when the mixing effect is strong."

The actual volume may be estimated by using an anemometer, or may be closely calculated from a flue gas analysis by using the following formula.

$$A = 11.6\left[\frac{CO_2 + \frac{CO}{2} + O_2}{CO_2 + CO} \times C + 3\left(H - \frac{O}{8}\right)\right]$$

where

A = weight of dry air per pound of dry coal;

CO_2, CO, and O_2 = per cent. volume of each in the flue gas;

C, H and O = the weight of each in 1 lb. of dry coal.

The total weight of the dry products of combustion passing out of the stack will be

$$W_1 = C \times \left(1 + 11.6\,\frac{CO_2 + \frac{CO}{2} + O_2}{CO_2 + CO}\right) + 26.8\left(H - \frac{O}{8}\right) + N$$

[1] Report of Coal Waste Commission, Pa., 1893, p. 31.

[2] Report of Coal Waste Commission, Pa., 1893, p. 31.

or, expressed in terms of "*A*," above

$$W_1 = A + C + N - 8\left(H - \frac{O}{8}\right)$$

where

W_1 = total weight of dry combustion products per pound of dry coal;
N = the weight of nitrogen in 1 lb. of dry coal.

The specific heat of the dry flue gas may be taken at 0.24 for ordinary purposes of calculation of B.t.u. loss.

The loss due to an incomplete combustion of the carbon to CO will be, in units of B.t.u. loss per pound of fuel,

$$\text{B.t.u. loss} = C \times \frac{10{,}160 \times CO}{CO_2 + CO}.$$

"The conclusions drawn by R. S. Hale[1] are: that ordinary firing is apt to give 10 to 20 per cent. worse results than the best skilled firing, the low results being caused by using too much air and by getting poor combustion.

That it is easier for firemen to get better results in some boiler furnaces than others, but that this difference becomes large only with poor soft coal.

That many but not all of the patent devices (down-draft grates, stokers, etc.) in common use will with moderately skilled firemen give better results than those obtained by ordinary firemen in ordinary furnaces.

That it is probable, but not proved, that ordinary firemen can get better results from these devices than can ordinary firemen on ordinary grates.

Heat of Combustion.—The heat produced by the combustion of 1 lb. of various substances is given in the following table in British thermal units:

Total Heats of Combustion

	B.t.u. per lb.
Hydrogen	62,032
Carbon to carbon dioxide	14,500
Carbon to carbon monoxide	4,400
Carbon monoxide to carbon dioxide	4,330
Olefiant gas	21,344
Liquid hydrocarbons vary in proportion to weight from	19,000
to	22,600
Charcoal, wood	13,500
Charcoal, peat	11,600
Wood, dry, average	7,800
Wood, 20 per cent. moisture	6,500
Peat, dry, average	9,950
Peat, 25 per cent. moisture	7,000
Coal, anthracite, best qualities, about	15,000
Coal, anthracite, ordinary, about	13,000
Coal, bituminous, dry, about	14,000
Coal, cannel, about	15,000
Coal, ordinary poor grades, about	10,000

These figures are slightly altered by different authors. The above list may fairly be taken as an average."

[1] Steam Users' *Circular*, No. 9.

Boiler Rating.—It has been customary in the past to rate fire-tube boilers at 11.5 sq. ft. of heating surface per horsepower, water-tube boilers at 10 sq. ft. of heating surface, and internally fired boilers at 8.5 sq. ft. of heating surface per horsepower, a horsepower being understood to be the "Centennial rating," 34½ lb. of water evaporated per hour from and at 212°F. This rating is in use today, but most engineers buy square feet of heating surface and not horsepower, this being due to the great advances which have been made in the art of firing boilers. At the time of the formulation of the "Centennial rating" 3.5 lb. of evaporation per square foot of surface per hour was considered very good work and the economical point of evaporation. At the present time with better designed boilers and our better knowledge of combustion problems it is not uncommon to obtain 9 or 10 lb. of evaporation, with good economy, from large-tube boilers and as much as 15 to 18 lb. with the smaller tube types. Prof. Bone, with his surface combustion boiler, has evaporated from 30 to 45 lb. of water per square foot of surface per hour over a considerable time, with excellent economy.

Considerable discussion has arisen regarding the measurement of heating surface, *i.e.*, whether such measurements should be based on the inside or outside diameters of the tubes.

The American Society of Mechanical Engineers favors using the surfaces which receive the heat—the outside diameter of water-tube and the inside diameter of fire-tube boilers.

Boiler Efficiency.—Boiler efficiency may be graded as follows: 50 to 60 per cent. poor; 60 to 70 per cent. fair; 70 to 75 per cent. good; over 75 per cent. excellent. The last is seldom obtained.

Pounds of Water per Horsepower-hour.—James Watt's figure was 1 cu. ft., or 62.5 lb. The standard of 1876, Centennial, adopted by the American Society of Mechanical Engineers was 30 lb. evaporated from a temperature of 100°F. (feed water) into steam at 70 lb. gage pressure.

This is equivalent to 34.5 lb. from and at 212°F.

As has already been pointed out there is no direct relation between boiler and engine horsepower.

For convenience in comparing the evaporative results of boilers operating under different conditions of feed-water temperature and steam pressure, it is necessary to reduce all such variable conditions to a definite standard. The conditions agreed upon, which have been in use many years, are those of a feed-water temperature of 212°F. and the evaporation of the water at that temperature into steam at atmospheric pressure, with a temperature of 212°F. This has been shortened into either "Equivalent evaporation" or "Evaporation from and at 212°."

Factor of Evaporation.—At the pressure of one atmosphere (14.7 lb. per square inch) and at 212°F., the heat necessary to make water at that temperature into steam at that pressure is approximately 970 B.t.u.

If, then, the total heat, Q, required to vaporize a weight of water, W, be observed from a test, in which the feed water was introduced at t', and the evaporation took place into steam at t, the total heat which went into the evaporated water was the product $Q \times W$.

If the evaporation had taken place from and at 212°F. Q would have been 970 for each pound, so that $970H$ would have been the equivalent heat absorbed if H is the corresponding weight of water evaporated from and at 212°F.

Equating these,

$$QW = 970H \text{ or } H = \frac{QW}{970}$$

gives the pounds of water which would have been evaporated from and at 212°.

A table giving the value of the factor $\frac{Q}{970}$ has been computed and may be found in various books dealing with boiler tests. This factor is designated, "Factor of Evaporation."

For example, given feed water at 40°F. evaporated into steam at 100 lb. gage, what is the factor of evaporation?

If q = the heat of the liquid at 40°F. = 8.1
q_1 = the heat of the liquid at 100 lb. gage (338°F.) = 309
r = the heat of vaporization at 100 lb. gage = 879.5
then $q_1 - q = 309 - 8.1 = 301$
then total heat = 301 + 879.5 = 1,180.5 B.t.u. per pound
$\frac{1,180.5}{970} = 1.22$ = factor of evaporation, that is, the evaporation of 1 lb. of water from a feed-water temperature of 40°F. into steam at 100 lb. gage pressure is equivalent to evaporating 1.22 lb. from water at 212°F. into steam at 212°F.

The average factor of evaporation for the wide range of feed-water temperatures and steam pressures in common use is roughly 1.15. This approximate factor may be used for all general calculations that do not require close refinement. This corresponds to the evaporation of 30 lb. of water per horsepower-hour under ordinary commercial conditions as $\frac{34.5}{1.15} = 30$.

Pounds of Water Evaporated per Pound of Dry Coal (from and at 212°).—

Maximum theoretical	15
Maximum under conditions of practice	12
Excellent practice	10
Fair practice	8
Common practice (small plants)	7

Pounds of Coal per Square Foot of Grate Area per Hour.—

(*a*) With chimney draft:

Slowest rate, Cornish boilers	4– 6
Ordinary rate, Cornish boilers	10– 15
Ordinary rate for anthracite	15– 20
Ordinary rate for bituminous	20– 25

(*b*) With forced draft:

Stationary water-tube boilers	25– 50
Locomotives	40–100
Torpedo boats	60–125

Boiler Deterioration.—Boilers are subject to many deteriorat forces which may be summed up as:

(*a*) Internal corrosion.
(*b*) External corrosion.
(*c*) Pitting.
(*d*) Grooving.
(*e*) General wear and tear.

Idle Boilers.—When boilers are out of service for any length of ti they should receive the following treatment as specified by Parsons.

"The outside should be cleaned and painted with a good metallic pa applied directly to the cleaned and dried surface. If the boiler be covered lagging, the lagging should not be allowed to absorb moisture from the atmosph

On the fire side, the soot and ashes should be thoroughly removed and surface cleaned. These surfaces should then be kept dry and not exposec damp air. Fresh lime in pans or trays, renewed as required, will absorb moisture in the air. Occasional small fires of tarred wood will be beneficial the heat will dry the metallic surfaces and the resinous condensations from thick smoke will cover the tubes and shell with a protective coating.

On the water side, corrosion may be active at the water line if the boilei left partly full. Idle boilers should, therefore, be entirely dry or complet filled with water. If the laying off is a short time only, it is a good plan to the boiler with water made alkaline by a little soda. If for a long period seems best to empty the boiler and dry out the inside by a small fire built i pan, which can be inserted through the lowest manhole. The manhole and ha hole covers can be put back and the boiler made tight so that the oxygen will consumed by the fire, or the covers can be left off and lime in trays used to abs any moisture.

Boiler Explosions.—Parsons states that explosions occur when steam pressure exceeds the resisting strength of the metal structure.

In a well-designed boiler the parts are of approximately equal stren throughout. It is good practice so to design a boiler that those pa shall have an excess of strength which are expected to suffer most rapi from corrosion or wear and tear. Then as the boiler advances in a the various parts become more nearly equal in strength.

Should a boiler become weakened and a rent occur, the steam pressure will be suddenly reduced, thus releasing the heat stored in the water. A portion of the water instantly flashing into vapor probably accounts for the great destructive effects produced by an explosion.

While the rent primarily occurs at some weak spot, the fracture may not and seldom does follow a line of structural weakness. The new forces set up at the instant of explosion no doubt account for this phenomenon.

All things being equal, the damaging effect by explosion of water-tubular boilers will be less than of fire-tubular boilers of equal rating, since the former contain a smaller proportion of water, and since extra time will be required for complete release, because the bursting part is small.

Failures of boilers are usually due to wear and tear, produced chiefly by expansion and contraction, to corrosion, to overheating and to carelessness. Overheating may be caused by low water or by scale or grease. Important fixtures, such as main stop valves, may become attacked, or the main steam pipe may be burst by water-hammer, thus causing a sudden release of pressure, which, if quick enough, may be followed by an explosion.

When an explosion does occur, it is frequently difficult to determine the cause, and hasty judgment should always be withheld. A good piece of metal may show a poor quality of fracture on account of the suddenness of the rupture. Opinion as to the quality of the metal should only be given after a close and careful analysis of physical and chemical tests.

The best way to prevent explosion is to employ intelligent labor and not neglect proper and regular inspection."

Boiler Inspection.—The requirements and regulations regarding inspection are given in the American Society of Mechanical Engineers' Code.

Number of Boilers to do Given Work.—The subdivision of heating surface into the proper number of boilers is important, for a careful study may result in much saving in first cost and in cost of operation.

For instance, if boiler capacity to evaporate 33,600 lb. of water per hour from and at 212° is required, approximately 9,600 sq. ft. of heating surface will be needed.

If each square foot of heating surface may be overloaded $33\frac{1}{3}$ per cent. (which is quite possible in ordinary practice) it is evident that if the 9,600 sq. ft. were divided among four boilers, one boiler might be shut down for repairs or cleaning, and the other three run at $33\frac{1}{3}$ per cent. overload and still evaporate 33,600 lb. of water.

If the total heating surface were divided into three boilers, each of 3,200 sq. ft. of heating surface, two might not be able to run the plant alone, so a fourth or spare boiler would have to be supplied. This would be poor division of power as the money spent on the spare boiler would represent so much capital lying idle most of the time.

Selection of Boiler Type.—The choice of type will depend much upon the conditions of service.

For high pressures such as are used for modern power plants, water-tube boilers are safer. They are also probably more economical and meet the varying demands better than fire-tube boilers.

For reasonably low pressures in relatively small power plants and for heating installations the ordinary fire-tube boiler meets the requirements well if overload capacity is not an essential factor. Such boilers are cheaper and probably cost less for repairs than water-tube boilers.

The tendency with fire-tube boilers is toward hand-fired furnaces, which are often objectionable because of excessive smoke production which may make them undesirable for urban conditions.

Saving by Use of Mechanical Stokers.—The difference between good and bad firing may easily amount to from 5 to 20 per cent. of the amount of fuel fired; hence, there is no investment around a steam plant which will pay better than the extra amount paid to secure good boiler practice.

Automatic stokers are now developed to a remarkable degree of perfection, and when suited to the fuel have an advantage over hand-firing in that under all conditions they are reliable, can be adjusted to the minimum of air and the maximum of load, and can be depended upon to operate continuously with the minimum amount of skilled labor. The economic saving will depend on the basis of comparison and the method of operation. Compared with an ordinary or poor fireman, they should show a large saving. Whether a stoker will save labor in the fire room depends upon the size of the plant. As a rule, mechanical stokers are not labor-saving devices in plants containing less than six to eight boilers (1,500 to 4,800 hp.).

One man can handle the coal and ashes, fire the boilers and attend to the water level of 200 hp. of boilers equipped with the common hand-fired furnace. With shaking or dumping grates 300 hp. may be controlled by one man. With large boilers equipped with dumping grates one man will fire around 1,000 boiler hp. when using the steam sizes of anthracite coal, but the coal must be delivered in front of the boiler and a water tender is usually provided for every 24 boilers. With soft coal about 700 boiler hp. may be fired by one man under similar cirumstances. In a large plant containing twelve 650 B. & W. boilers, equipped with stokers, a water tender, one fireman and one helper are required per watch for their efficient operation. In stations of this size the ash men are in the basement, and the change from hand- to stoker-firing would make no difference in their number.

One authority states that stokers save 30 to 40 per cent. of the boiler labor in plants using over 200 tons of coal per week; 20 to 30 per cent. in plants using from 50 to 200 tons of coal per week, and no saving in plants below 50 tons.

It should be remembered that unless the type of stoker is suited to the kind of fuel obtainable, the maintenance of the stoker plant is likely to be extremely high, running in some cases twice or three times as high as fire-room labor under hand-fired conditions.

Cost of Mechanical Stokers.—In general, mechanical stokers cost from $3.50 to $6.50 per boiler horsepower, but the cost depends more on the width of the stoker than on the horsepower of the boiler. Chain-grate stokers cost in the neighborhood of from $180 to $250 per foot of width. Inclined-grate stokers of the Roney, Acme, Wilkinson or similar types, from $140 to $225 per foot of width. Underfeed stokers from $200 to $300 per foot of width. The length of the stokers is usually standard and depends on the type of coal to be burned. These prices differ considerably with the amount of auxiliary material furnished with the stoker, such as fronts, air boxes, coking arches, stoker drives and speed-changing devices, but are based on labor and material costs current in New York prior to the European war.

The following is the approximate cost of stokers suitable for a water-tube boiler of 350-hp. rated capacity with 45 sq. ft. of grate surface; height of chimney above grate, 175 ft.; coal burned, Illinois screenings. The cost of the installation is not included.

1. Burke smokeless furnace	1,000
2. Wilkinson stoker	1,200
3. Roney stoker	1,300
4. Hawley down-draft furnace	1,350
5. Murphey furnace and stoker	1,350
6. Jones underfeed stoker	1,400
7. Chain grate and appurtenances	1,500
8. Taylor stoker	2,000

R. J. S. Pigott (*Proceedings* Am. Elec. Ry. Assoc., 1914) gives the following data for mechanical stokers.

AVERAGE DATA FOR STOKERS

Type of stoker	Step and slope overfeed	V over-feed	Chain over-feed	Gravity under-feed	Horizontal retort underfeed
Average price per rated boiler horsepower	$3.60	$3.60	$3.50–$6.55	$5.65	$4.44
Normal forcing ability in per cent. of rating	190	175	260	300–350	300
Price per maximum horsepower developable	$1.90	$2.06	$2.52	$1.62–$1.88	$1.48
Maintenance per ton coal fired, in cents	10–12	11–14	6–10	2.5–4	4–6
Attendance in man-hours per active hour	0.45	0.45–0.50	0.20–0.30	0.08–0.10	0.30–0.40
Pounds coal per square foot grate surface (maximum)	35–38	35–42	45–48	60–75	50–65

Operation and Care of Boilers.—Full instructions regarding the operation and care of steam boilers will be found in the Code published by The American Society of Mechanical Engineers (1916).

Among the most important points upon which the power plant engineer should be informed are:

(*a*) Water level.
(*b*) Leaks.
(*c*) Getting up steam.
(*d*) Cutting in boilers.
(*e*) Low water.
(*f*) Foaming.
(*g*) Safety valves.
(*h*) Blowing off.
(*i*) Grease.
(*j*) Efficient operation.
(*k*) Banking fires.
(*l*) Scale prevention.
(*m*) Shutting down.
(*n*) Inspection and repairs.
(*o*) Laying up boilers.

PROBLEMS

32. Given the following data from a boiler test:

1. Kind of boiler, Heine, water-tube.	
2. Kind of fuel, West Virginia, briquettes.	
3. Furnace, hand-fired.	
4. Duration of trial, hours	10.25
5. Grate surface, square feet	40.55
6. Water heating surface, square feet	2,031.0
7. Steam pressure, gage, pounds per square inch	83.7
8. Temperature of feed water, °F	52.9
9. Temperature of escaping gases from boiler, °F	590.0
10. Total weight of coal as fired, pounds	7,515.0
11. Moisture in coal, per cent	2.32
12. Ash and refuse in dry coal, per cent	10.36
13. Calorific value per pound of dry coal, B.t.u	15,235.0
14. Calorific value per pound of combustible, B.t.u	16,266.0
15. Moisture in steam, per cent	0.8
16. Total weight of water fed to boiler, pounds	62,641.0
17. Factor of evaporation	1.20

Find the following:

1. Ratio of water heating surface to grate surface.
2. Total weight of dry coal consumed, pounds.
3. Total ash and refuse, pounds.
4. Total combustible consumed, pounds.
5. Dry coal consumed per hour, pounds.
6. Combustible consumes per hour, pounds.
7. Dry coal per square foot of grate surface per hour, pounds.
8. Quality of steam (dry steam = unity).
9. Water actually evaporated, corrected for quality of steam, pounds.
10. Water evaporated per hour, corrected for quality of steam, pounds.
11. Equivalent evaporation per hour from and at 212°, pounds.
12. Equivalent evaporation per hour from and at 212°, per square foot of water heating surface, pounds.
13. Horsepower developed.

14. Builders' rated horsepower.
15. Percentage of builders' rated horsepower developed.
16. Water evaporated under actual conditions per pound of coal as fired.
17. Equivalent evaporation from and at 212° per pound of coal as fired.
18. Equivalent evaporation from and at 212° per pound of combustible.
19. Equivalent evaporation from and at 212° per pound of dry coal.
20. Efficiency of boiler; heat absorbed by boiler per pound of combustible, divided by the heat value of 1 lb. of combustible, per cent.
21. Efficiency of boiler and grate; heat absorbed by boiler per pound of dry coal, divided by heat value of 1 lb. of dry coal, per cent.

33. Given the following data from a boiler trial:

1.	Heine water-tube boiler.	
2.	Iowa coal.	
3.	Duration of trial in hours	9.92
4.	Grate surface, square feet	40.55
5.	Water heating surface, square feet	2,031.0
6.	Steam pressure, gage, pounds per square inch	82.5
7.	Temperature of feed water, °F.	48.0
8.	Temperature of flue gases, °F.	627.0
9.	Total weight of coal as fired, pounds	10,986.0
10.	Moisture in coal, per cent.	14.88
11.	Ash and refuse in dry coal, per cent.	17.4
12.	B.t.u. per pound of dry coal	11,497.0
13.	B.t.u. per pound of combustible	13,385.0
14.	Moisture in steam, per cent.	0.91
15.	Total weight of water fed to boiler, pounds	55,180.0
16.	Factor of evaporation	1.205

Determine the values indicated in problem 32.

34. An office building contains 7,500 sq. ft. of radiation for steam heating, supplied from a low-pressure fire-tube boiler of 950 sq. ft. of heating surface. The engine used for power purposes, running non-condensing and exhausting into the atmosphere consumed in an 8-hr. run 27,700 lb. of steam supplied from a water-tube boiler of 950 sq. ft. of heating surface. What boiler horsepower was being developed by each boiler? What per cent. of the manufacturer's rating was developed in each case?

How much coal was probably used in the 8 hr. run for all purposes if the coal contained 5 per cent. moisture? What was the consumption of the engine per indicated horsepower-hour if the switchboard readings were 240 volts and 260 amperes? D.C. generator.

35. What are the approximate boiler efficiencies corresponding to the table of equivalent evaporations per pound of dry coal on page 157.

36. Find the (a) factor of evaporation; (b) the equivalent evaporation; (c) the B.t.u. output of boiler per hour, and (d) the boiler horsepower required for each of the following installations.

1. A heating system using 2,200 lb. of steam per hour, the steam being delivered from the boiler under 5 lb. per square inch gage pressure and at 96 per cent. quality, and the condensate being returned to the boiler at 175°F.

2. A non-condensing steam engine carrying 150 i.hp. load, requiring with the auxiliaries 29 lb. of steam per indicated horsepower-hour; steam pressure 125 lb. per square inch gage; quality 98.9 per cent.; feed-water temperature 110°F.

3. A 500-kw. steam turbine, requiring with auxiliaries 18 lb. of steam per kilowatt hour; steam pressure 160 lb. per square inch gage; superheat 100°F.; feed-water temperature 210°F.

37. A boiler plant consisting of three 250-hp. hand-fired boilers uses No. 3 buckwheat anthracite coal, costing $2.50 per long ton as delivered. Twelve hundred tons are used per month at an average equivalent evaporation of 7 lb. per pound of coal as fired. The operating labor is in three shifts, each consisting of two firemen and one coal passer, paid $2.50 and $2, respectively, per day, 7 days per week. The per cent. of ash in the coal by analysis is 14, but the total ash and refuse are approximately 19 per cent., costing 40 cts. per ton for removal.

The use of soft coal and underfeed stokers is considered, the coal costing $3 per long ton delivered, the ash content being 8 per cent. An evaporation of 9 lb. is anticipated, the labor for operation being reduced to one fireman and one coal passer per 8-hr. shift, at the same wage rates. What, if any, will be the reduction in cost per 1,000 lb. of steam? On the basis of the same future demand for steam would the investment seem advisable?

38. If the above plant were to operate under the same load only 10 hr. per day (one shift, $2.50 and $2 per day wage rate), 5½ days per week, and if it had previously been using the $3 soft coal and obtaining an evaporation of 9 lb., would the stoker investment still be justified?

39. Coal of the following analysis is being used in a hand-fired furnace.

	Per cent. by weight
Carbon	70.5
Hydrogen	4.9
Nitrogen	1.8
Oxygen	8.2
Sulphur	0.9
Ash	13.7
	100.0
B.t.u. per pound dry	12,750

Analysis of the flue gas gives the following results:

	Per cent. by volume
CO_2	7.6
O	11.9
CO	0.3
N	80.2

Determine:

(*a*) The pounds of air theoretically required for perfect combustion per pound of coal.

(*b*) The pounds of air actually supplied per pound of coal.

(*c*) The per cent. excess air.

40. On the basis of the analyses in problem 39, with a boiler-room temperature of 70°F. and a flue temperature of 580°F., how many B.t.u. are lost in the dry flue gases per pound of dry coal. What per cent. of the heat value of the coal is this heat loss?

41. After closing up the leaks in the boiler setting of problems 39 and 40 and adopting better methods of firing the following average flue gas analysis is obtained.

	Per cent. by volume
CO_2	13.1
O	6.5
CO	0.4
N	80.0
	100.0

If in both cases the losses other than the sensible heat in the flue gas are assumed as 16 per cent. of the heat in the coal, what would be the probable yearly saving in coal bill, coal costing $3.50 per ton; the former consumption having been 1,500 tons per year.

CHAPTER VII

CHIMNEYS AND MECHANICAL DRAFT

Chimneys.—Chimneys are built primarily for two purposes; first, to furnish draft to enable a sufficient quantity of combustible to be burnt, and second, to discharge hot or noxious gases at a sufficient height to avoid a nuisance.

Theoretically, the draft power of a chimney depends on the height of its top above the grate bars and the respective densities of the hot gases and the outside air.

Let H = height of the chimney above the grate in feet.
t_0 = 493°F. absolute temperature at 32°F.
t_1 = absolute temperature of outside air.
t_2 = absolute temperature of gases.
δ = theoretical draft power in inches of water.

$$\delta = 0.01549H\left(\frac{t_0}{t_1} - \frac{t_0}{t_2}\right)$$

This formula is based on the supposition that t_2 is the mean temperature of the hot gases in the stack. Assuming a mean stack temperature of 600°F. with the external air at 62°F. the above formula reduces to $\delta = 0.00736H$, on which the following table is based:

H	δ	H	δ	H	δ	H	δ
10	0.074	60	0.441	110	0.810	160	1.178
20	0.147	70	0.515	120	0.883	170	1.251
30	0.220	80	0.589	130	0.957	180	1.325
40	0.294	90	0.662	140	1.030	190	1.398
50	0.368	100	0 736	150	1.104	200	1.472

In practice, chimney height may be determined from the draft requirements by the following formula:

$$H = \frac{\delta}{0.00736} = 135.87\delta$$

The required draft power depends upon the loss of head due to friction in ashpit air admission openings, to friction in passing through grate and fuel bed, to losses by leakage of cold air into combustion chamber, to friction in the boiler passes and finally to flue, economizer and stack frictions and the difference of temperature necessary to produce the flow in the stack. Of these, the loss due to the grate and the fuel bed amounts to from 50 to 75 per cent. of the total. The loss from leakage of cold air into combustion chamber, from friction in the boiler passes, flues, economizer and stacks amounts to from 15 to 35 per cent., leaving often as little as 4 per cent. to produce velocity in the chimney gases.

No satisfactory method has been devised for calculating the necessary draft.

The height may also be determined from the desired fuel consumption per square foot of grate per hour.

Let F = pounds of coal burnt per hour per square foot of grate. Then following Thurston:

For anthracite coal

$$\delta = 0.001875(F - 1)^2 \text{ and } H = \frac{(F-1)^2}{4},$$

For best Penn. or Welsh

$$\delta = 0.00148F^2 \qquad \text{and } H = \frac{F^2}{5}.$$

For Pittsburgh or Illinois

$$\delta = 0.000833F^2 \qquad \text{and } H = \frac{F^2}{9}.$$

These formulas have only a limited application.

Natural draft greater than 1.5 in. of water is seldom necessary, and higher intensities can much better be obtained by forced or induced draft. This limits the height of chimneys to about 200 ft., which is perhaps above the economical limit from a cost and construction standpoint. Chemical and metallurgical works in the neighborhood of towns require excessively high chimneys to remove the noxious gases and a number have been built exceeding 400 ft. in height. In many works, however, means have been taken to utilize sulphurous, arsenical and other vapors with a large measure of success.

All chimney formulas are based on the hypothesis that the capacity or theoretical coal consumption of a chimney varies directly as the area (or effective area) and the square root of the height.

Let C = coal consumption in pounds per hour.
A = area of chimney in square feet.
H = height above the grates in feet.

Then the typical formula may be written thus:

$$C = KA\sqrt{H},$$

where K = a constant. It may be written

$$C = K(A - 0.6\sqrt{A})\ \sqrt{H} \text{ (Kent's formula),}$$

where $(A - 0.6\sqrt{A})$ is the "effective area."

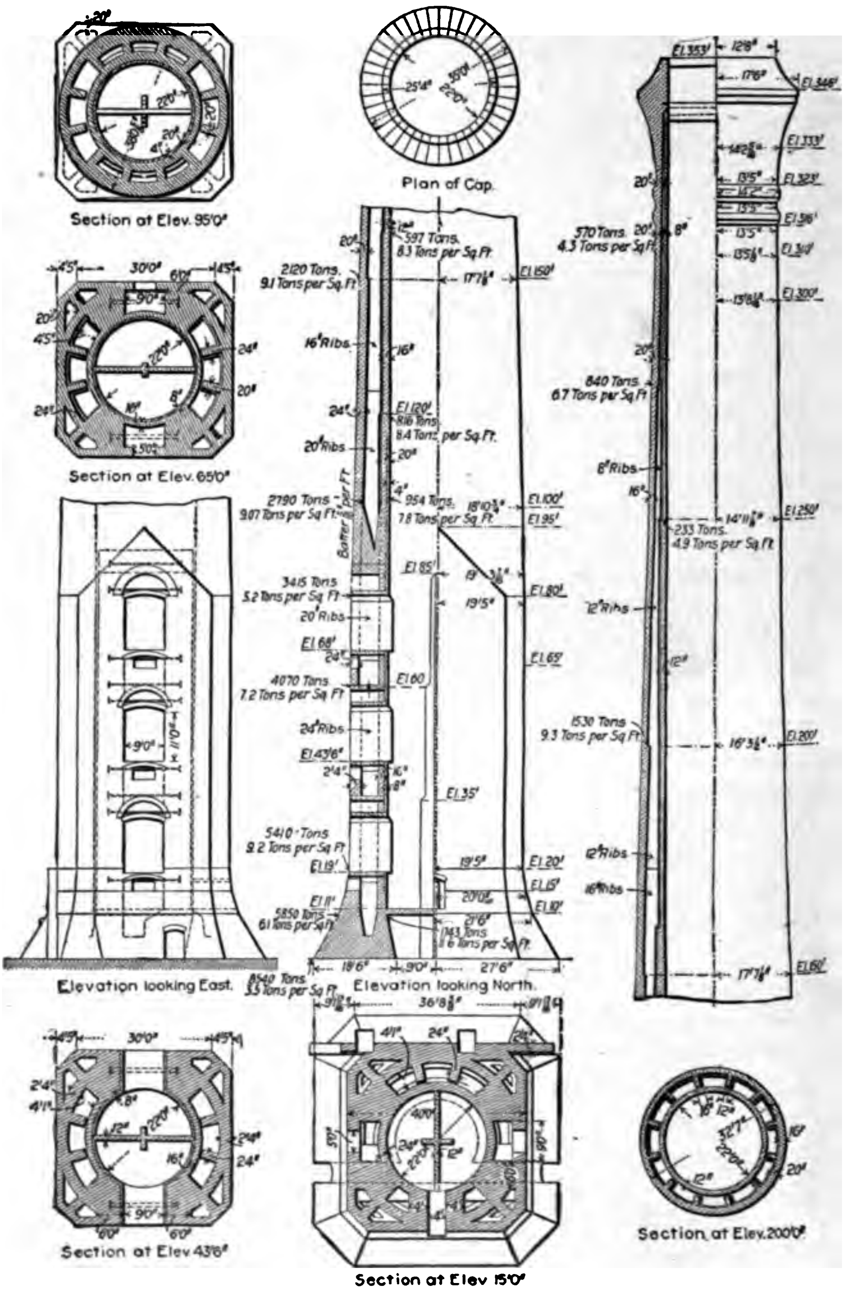


FIG. 107.—Common brick chimney, 96th St. Power House, Metropolitan St. Ry. Co New York.

The value of the constant K, as given by different authorities, varies greatly. Toldt gives $K = 5$; Prechtl, $K = 6.4$; Molesworth, $K = 9$; Ser, $K = 9.3$; Hutton, $K = 10$ to 16; Seaton and Rounthwaite, $K = 12$; Henthorn, $K = 16.6$, and Kent, $K = 16.65$ (using the effective area for A); Brinckerhoff (average), $K = 18.1$. Toldt and Prechtl refer mainly to German metallurgical practice, Ser to general French practice, Molesworth and Hutton to English practice, Seaton and Rounthwaite to marine practice, Henthorn to American mill practice.

An average of 30 stacks of various sizes now doing good work gives $K = 9.4$. An average of three notoriously overworked stacks gives $K = 17.9$.

Ser's figure $K = 9.3$, was obtained theoretically by allowing for twice the amount of air necessary for perfect combustion. By allowing an excess of one-half the amount necessary for perfect combustion, which result can readily be obtained by the use of automatic stokers, the constant $K = 12$ will be obtained.

For preliminary calculations the above formula with $K = 12$, gives practical results, but the chimney should be checked by comparison with known stacks of similar diameter and height for the final calculations.

The value of $K = 12$ applies only to brick-lined stacks. In case an unlined iron or steel stack is being considered, the value of K may be increased to 14 or 15, and for small stacks 16 may be used.

The base of a brick stack should rarely be less than one-tenth of the height. The allowable batter according to different authorities varies from 1 in 192 to 1 in 20 on each side, but the best practice lies between 1 in 30 and 1 in 40 for ordinary brick, with 1 in 60 to 1 in 80 for the Custodis or hollow-tile method of construction.

In brick chimneys practice varies as to the thickness of the walls. The linings are not exposed to wind pressure, and consequently can be much thinner than the outside wall, but 100 ft. is about the practical limit of each step. The usual practice is to make the steps about 50 ft. high and 4 in. thick at the top up to a height of 150 ft.

For higher chimneys the lining should be 8 in. thick at the top. The outside walls for chimneys up to 150 ft. high may be 8 in. thick at the top, with the steps about 50 ft. high or the upper steps may be as high as 60 ft., with 50 ft. for the lower steps. For stacks built on the Custodis principle the top courses are from 8 to 13 in. thick, depending on the height. The thickness of the moulded brick is increased 2 in. every 5 meters, or about every 16½ ft.

All brick stacks should be topped with a waterproof cap, usually of cast iron, although in many cases it is made of stone or monolithic concrete. Lightning rods are considered by many engineers as a necessity, and there is no doubt that many stacks have been saved by their

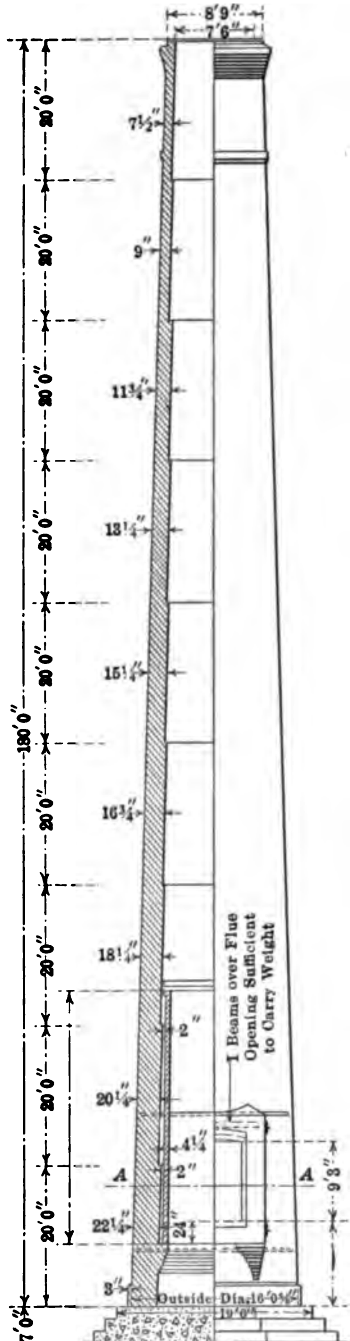


FIG. 108.—Typical hollow-tile chimney.

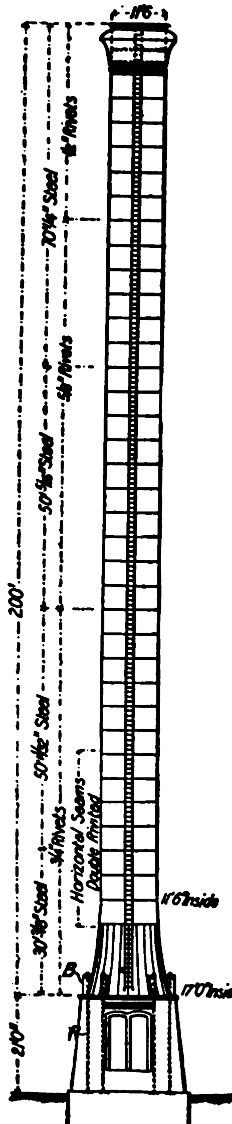


FIG. 109.—Steel stack, Wilmerding, Pa.

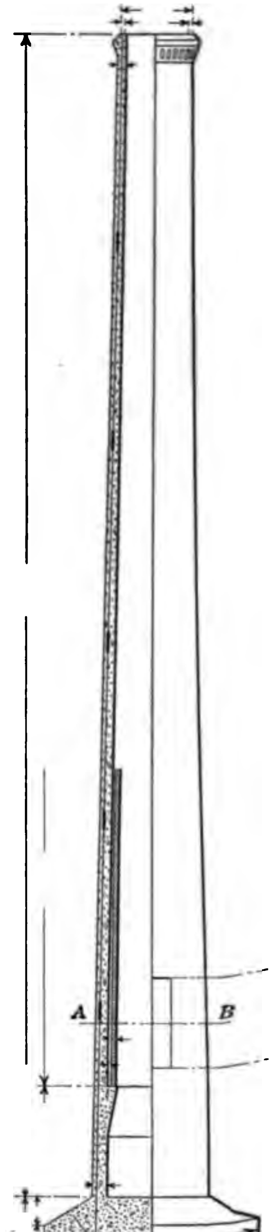


FIG. 110.—Tapered reinforced-concrete chimney M. W. Kellog Co.

use. If furnished at all, care should be taken that the conductor and ground connections are good.

The best constructions for steel stacks include a number of vertical stiffeners riveted to the shell which support horizontal cast-iron or steel rings on which the linings are built. The vertical stiffeners are usually spaced about 5 ft. apart, and the horizontal rings about 20 ft. apart. By this method any section of the lining may be replaced without disturbing the other sections. The thickness of the metal at the top of the stack in such cases is usually 3/8 in., increasing 1/8 in. every 50 ft. Stacks in which the linings are not supported may be 1/4 in. thick at the top, increasing 1/16 in. every 30 ft.

Guyed stacks of light sheet iron are frequently used for single boilers and even for quite large plants especially where the expected life of the plant is short. The smaller stacks are made up in lengths of about 20 ft. of 1/8-in. steel connected by angle rings on the outside or the whole stack may be riveted in one piece on the ground and erected with a gin pole.

For these stacks the value of K in the general formula may be as high as 20 as they are usually connected directly to the boiler uptake and are exposed to high temperatures.

Such stacks deteriorate very rapidly and cannot be considered a desirable construction, but occasionally circumstances will require their use. Galvanized stranded wire cables form the best guys and the anchors may be concrete blocks for the larger sizes. For the smaller sizes the guys usually lead to the steel building structure.

During the last 15 years the use of reinforced concrete as a stack material has become quite common. These stacks are of many patterns and have been quite successful. The later stacks resemble the brick stacks on the outside, but cylindrical and bottle shapes are used to some extent.

The advantages claimed for this type of stack are.:

1. Absence of joints, the construction being monolithic.
2. Rapidity of construction.
3. Great strength in compression and tension.
4. Light weight, requiring little foundation.

Disadvantages.—

1. Difficulties with forms.
2. The break at the end of each day's work.
3. No data concerning life available.

Many good stacks of this type have been erected in the last few years and their cost, appearance and performance compares favorably with the other types.

Evasé Stacks.—During the last few years a type of stack has been developed in Europe which offers marked advantages both as to cost and ease with which the draft may be controlled. The stack action is based on the injector principle, but the theory has not been well worked out as yet. The stack resembles a Venturi meter set up on end, the upper cone or diffuser enclosing an angle of 7°. These stacks are rarely over 60 or 70 ft. in height and are usually applied to single boilers or batteries, in order that the control may be perfect. It is usually possible to attain an evaporation of about 2 lb. of water per square foot of boiler surface with the stack alone. For the higher ratings air is injected just below the Venturi throat, thereby inducing a higher rate of suction than the height of the stack would make. It is possible so to proportion the stack and blower capacities that a suction draft of 3 or 4 ins. of water may be obtained, but this is usually unnecessary, as drafts of from 1 in. to 1½ in. will fulfil most of the requirements.

The following empirical rules may be followed in the tentative design of these stacks:

1. Figure the area and diameter of the base of the stack from the maximum number of cubic feet of flue gases per second, using 40 ft. per second as the velocity.
2. Make the area of the throat 50 per cent. of the stack area.
3. Figure the height of the suction cone (30° included angle).
4. The height of the diffuser will be seven times the throat diameter and the diameter of the stack at the top of the diffuser will be 1.85 times the throat diameter.
5. The next thing to settle is the size of the air nozzle for inducing draft. This is a matter of the static pressure of the available fans and is smaller as this static pressure is larger. It is usually taken in the neighborhood of 15 in. of water. The formula for the diameter ratios then becomes

$$\frac{r}{R} = 1.11 \sqrt{\frac{\text{suction pressure}}{\text{motive air pressure}}}.$$

Both pressures in inches of water. From this ratio the diameter and area of the nozzle may be readily calculated.

6. The amount of motive fluid must next be calculated, using $w = 0.9$ area $\sqrt{2g \times 70 \times \text{water gage}}$ of motive air. Knowing the cubic feet of motive air per second and the area of the nozzle, its velocity can readily be calculated, also the velocity head, which added to the static head gives the total head furnished by the induced-draft fan, whose horsepower can then be calculated by the following formula

$$\text{hp.} = \frac{h_t A V \times 60}{6{,}395 \times y}$$

where y is the fan efficiency, usually 0.50.

h_t is the total pressure, inches of water.

A is area of outlet.

V is velocity in feet per second.

These stacks have been used in a large number of the best modern stations in Europe, South America and also in the Rand in Africa.

Flues and Uptakes.—The uptakes on standard boilers are usually gned for a normal evaporation of 3½ lb. of water per square foot of ace per hour. This, ordinarily with soft coal, corresponds to ⅓ lb. oal per square foot of heating surface per hour or 133 cu. ft. of flue s per hour per square foot of heating surface. Taking the velocity

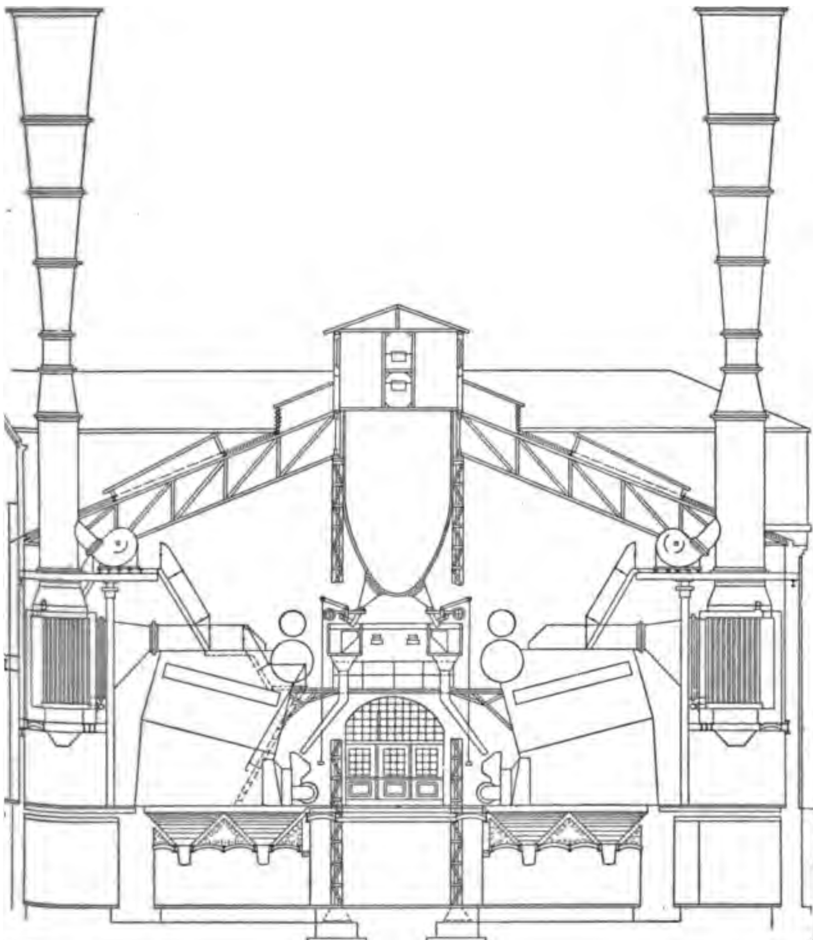


FIG. 111.—Section of boiler house with Evasé stacks.

the gas at rating to be 10 ft. per second this would correspond to a flue ea of 0.0037 sq. ft. for every square foot of heating surface. With a per cent. overload on the boiler this will give sufficiently low velocities make sure that the minimum portion of dust will be carried up the ck. Where soft coal is used this area may be safely reduced to 0.003

per square foot of heating surface, in fact, on some installations as low a value as 0.0025 has been used with success.

From the beginning of the uptake to the base of the stack the temperature of the flue gas continually decreases and it is good practice to take into account the drop in temperature and also to consider a 25 per cent. increase in velocity in the same space. Where increasing again in the stack the velocities are approximately:

	Normal	Overload
Velocity at uptake	10 to 15 ft.	15 to 20 ft.
Velocity at end of flue	13 to 18 ft.	18 to 23 ft.
Main velocity in stack	20 to 30 ft.	25 to 40 ft.

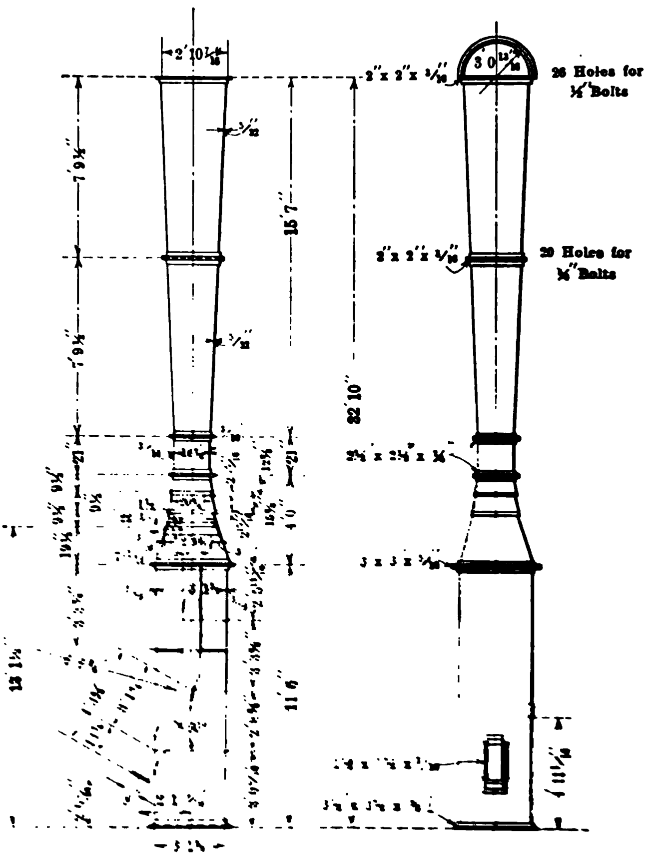


FIG. 112.—Evase stack.

These figures will be modified more or less due to variable amounts of excess air present in the flue gas.

Flues and uptakes should be as straight as possible and of as large an area as is consistent with the general design of the station. Sharp right-angled bends should be avoided and if possible the bottoms of the flues should be semicircular in section. The main portions of the flue for permanent work should be built of steel ¼ in., or thicker, the best practice being around ⅜ in., well stiffened with longitudinal angles at corners and cross angles or tees on the outside. No rivets should be used in the building up of the flue but square-headed bolts with square-headed nuts may be used. Where changes occur, plates should be bent wherever possible avoiding sudden contractions and expansions.

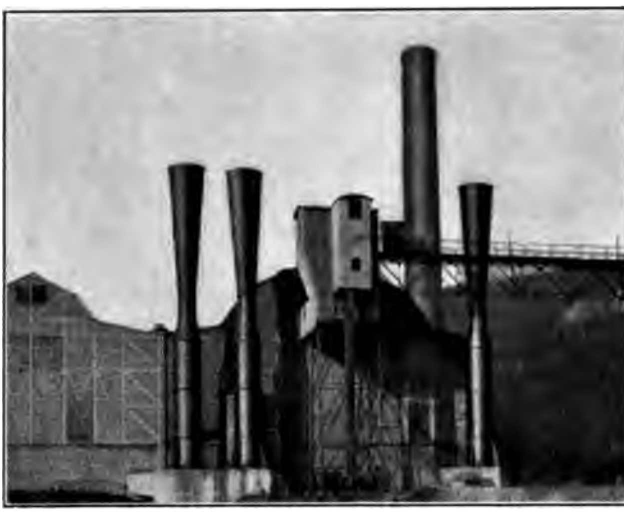


FIG. 113.—Evasé stacks at a German plant.

No paint should be used on the interior of the flue. The best protective covering for this purpose is a wash of Portland cement and water mixed to a consistency of cold-water paint and applied with a stiff brush. The outside of the flue should be covered with 2 in. of asbestos or magnesia covering fastened onto wire mesh, which has been wired securely to the stiffening angles, the usual construction leaving an inch air space between the flue and the covering.

Suitable expansion joints must be provided if the flue is of any great length.

The methods of hanging the flue deserve careful attention, the best way being to support it by steel straps from I-beams properly spaced and located just above the top plate of the flue. Suitable clean-out doors should be provided and pipes or chutes through which the flue dust may be taken away without being scattered over the neighboring machinery.

Chimney Dimensions.—The following table of chimney dimensions may serve as a guide in checking dimensions determined by the given formulæ.

TABLE OF CHIMNEY DIMENSIONS

Diameter in inches	Height, in feet 75	80	85	90	95	100	110	120	130	140	150	175	200
	Commercial horsepower												
24	75	78	81										
26	90	92	95	98									
28	..	106	110	114	117	120							
30	..	122	127	130	133	137	...						
32	..	...	144	149	152	156	164						
34	..	...	162	168	171	176	185						
36	..	...	...	188	192	198	208	215					
40	..	...	...	...	237	244	257	267	279				
44	..	...	...	...	287	296	310	322	337				
48	..	...	...	...	...	352	370	384	400	413			
54	..	...	...	...	...	445	468	484	507	526			
60	..	...	...	...	...	...	577	600	627	650	672		
66	..	...	...	...	...	...	697	725	758	784	815		
72	..	...	...	...	...	...	...	862	902	932	969	1,044	
84	..	...	...	...	...	...	...	1,173	1,229	1,270	1,319	1,422	
96	..	...	...	...	...	...	...	.....	1,584	1,660	1,725	1,859	1,983
108	..	...	...	...	...	...	...	.....	2,058	2,102	2,181	2,352	2,511
120	..	...	...	...	...	...	...	.....	.....	2,596	2,693	2,904	3,100

DRAFT PRESSURE REQUIRED FOR COMBUSTION OF DIFFERENT FUELS

Kind of fuel	Total draft in inches of water	Kind of fuel	Total draft in inches of water
Straw	0.20	Slack, very small	0.7–1.1
Wood	0.30	Coal dust	0.8–1.1
Sawdust	0.35	Semi-anthracite coal	0.9–1.2
Peat, light	0.4	Mixture of breese and slack	1.0–1.3
Peat, heavy	0.5	Anthracite, round	1.2–1.4
Sawdust mixed with small coal	0.6	Mixture of breese and coal dust	1.2–1.5
Steam coal, round	0.4–0.7	Anthracite slack	1.3–1.8

Cost of Guyed Iron Stacks.—

Approx. hp.	Height, ft.	Diam., in.	Price complete
25	40	16	$60
...	40	18	70
...	50	18	85
75	50	20	90
...	50	26	105
	60	22	110
100	60	24	125
	60	26	135
...	60	28	150
125	60	28	190
...	60	32	205
150	60	34	165
200	60	36	215
225	60	38	230
250	60	42	260
300	60	46	290
400	60	52	340
	100	60	500

Cost of Brick Chimneys.—

Approx. hp.	Height, ft.	Diam. flue	Square base	Outside wall		Cost fire-brick lining ½ height	Cost concrete fdtn.	Total cost
				No. brick	Cost at $14 per M			
85	80	25″	7′ 5″	32,000	$448	$60	$90	$598
135	90	30″	8′ 3″	40,000	560	82	144	786
200	100	35″	9′ 10″	65,000	910	113	198	1,226
300	110	43″	10′ 2″	75,000	1,050	190	252	1,492
400	120	51″	11′ 2″	87,000	1,218	261	306	1,785
750	130	61″	12′ 6″	131,000	1,834	334	360	2,528
1,000	140	74″	13′ 11″	151,000	2,114	432	414	3,060
1,650	150	88″	15′ 1″	200,000	2,800	482	468	3,750
2,500	160	110″	17′ 10″	275,000	3,850	720	525	5,095

The following approximate costs of various sizes of a well-known radial brick chimney give an idea of the variation in cost due to increase in diameter and height.

Size of chimney		Cost	Size of chimney		Cost
Height, ft.	Diameter, ft.		Height, ft.	Diameter, ft.	
75	4	$1,350	175	8	$7,050
75	6	1,950	175	10	7,525
75	8	2,650	175	12	8,050
75	10	3,725	175	14	9,725
....	...	.....	181	21	11,500
125	6	3,500	200	6	9,250
125	8	4,250	200	10	10,500
125	9	3,345	200	11	7,990
125	10	4,675	200	12	11,100
125	12	5,125	200	14	12,500
150	8	6,150	250	10	16,500
150	10	4,350	250	12	18,250
150	10	7,125	250	14	21,500
150	12	7,750	250	16	20,000
150	14	8,275	250	16	24,250

Cost of Special Chimneys.—Christie ("Chimney Design and Theory") gives the following cost of chimneys 150 ft. high and 8 ft. internal diameter.

	Approximate cost
Common red brick	$8,500
Radial brick	6,800
Steel, self-supporting, full lined	8,300
Steel, self-supporting, half lined	7,800
Steel, self-supporting, unlined	5,820
Steel, guyed	4,000

Average Cost of Stacks and Flues.—The average cost of stacks and flues (erected) for several installations ranging from 10 hp. to 2,000 hp. is reported by one consulting engineer as follows:

Cost of Stacks and Fuels per Engine Horsepower

Simple non-condensing:						
Engine horsepower	10	12	14	15	20	
Cost of stack, per horsepower	$16.00	$14.80	$13.40	$13.00	$11.60	
Cost of flues, per horsepower	2.30	2.30	2.30	2.30	2.25	
Engine horsepower	30	40	50	75		
Cost of stack, per horsepower	$8.70	$7.30	$6.30	$5.60		
Cost of flues, per horsepower	2.20	2.15	2.10	2.00		
Simple condensing:						
Engine horsepower	10	12	14	15	20	30
Cost of stack, per horsepower	$12.00	$10.70	$9.70	$9.40	$8.50	$6.30
Cost of flues, per horsepower	2.30	2.30	2.30	2.30	2.20	2.15
Engine horsepower	40	50	75	100		
Cost of stack, per horsepower	$5.70	$5.25	$4.80	$4.55		
Cost of flues, per horsepower	2.10	2.05	2.05	2.00		

COST OF STACKS AND FUELS PER ENGINE HORSEPOWER.—(*Continued*)

Compound condensing:						
Engine horsepower	100	200	300	400	500	600
Cost of stack, per horsepower	$4.55	$4.00	$3.65	$3.30	$3.10	$2.95
Cost of flues, per horsepower	1.95	1.80	1.60	1.35	1.10	1.00
Engine horsepower	700	800	900	1,000	1,500	2,000
Cost of stack, per horsepower	$2.90	$2.85	$2.80	$2.75	$2.70	$2.70
Cost of flues, per horsepower	0.90	0.80	0.70	0.55	0.55	0.55

Forced and Induced Draft.—In the ordinary power plant the chimney furnishes the draft necessary to burn the fuel. Systems of forced and induced draft have, however, been developed where it has been necessary to burn more coal, or where because of other difficulties, a better command was needed over the draft than could be obtained by a chimney. In the forced-draft system the stack is allowed to carry away the products of combustion, but the air necessary for the combustion of the fuel is forced under the grate by means of some type of blower so located as to

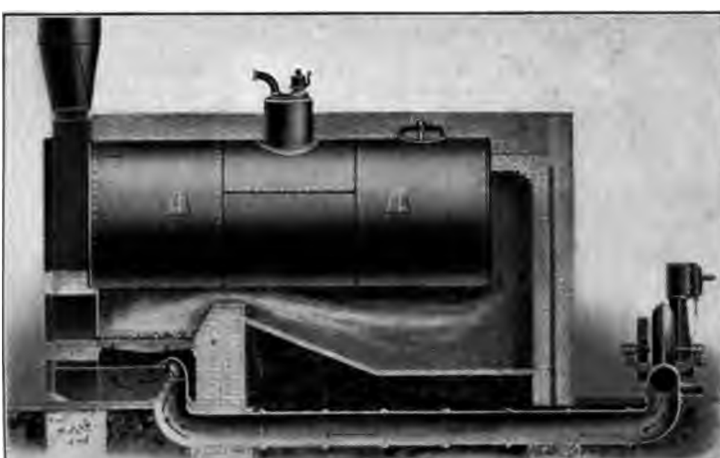


FIG. 114.—Forced draft.

deliver into a closed ashpit or into an air-tight fire room. In the induced-draft system a much larger fan, working at a lower pressure, is introduced into the flue, leading from the boilers to the stack, and the draft pressure of the stack is augmented by the pressure developed by the fan. Of the two systems the forced-draft system is most used, since the fan in the induced-draft system is particularly liable to deterioration on account of its being exposed to the action of the hot chimney gases. It was formerly claimed by the advocates of induced draft that the expense of a chimney could be saved by the adoption of this system, and this is true in some few cases. In the large majority of cases, however, the plant is so located that a stack of considerably greater height than would be considered necessary for draft alone must be provided to carry the noxious gases and smoke a sufficient distance above neighboring structures. Forced draft came into prominence through the use of the finer steam sizes of anthracite for fuel, as it was found that these sizes of anthracite could not be burned with any degree of satisfaction with a draft above the fire-bed. Some of the more successful of the modern types of stokers require forced draft for the burning of bituminous coal, and the ease of manipulation of a fire with the combination of forced

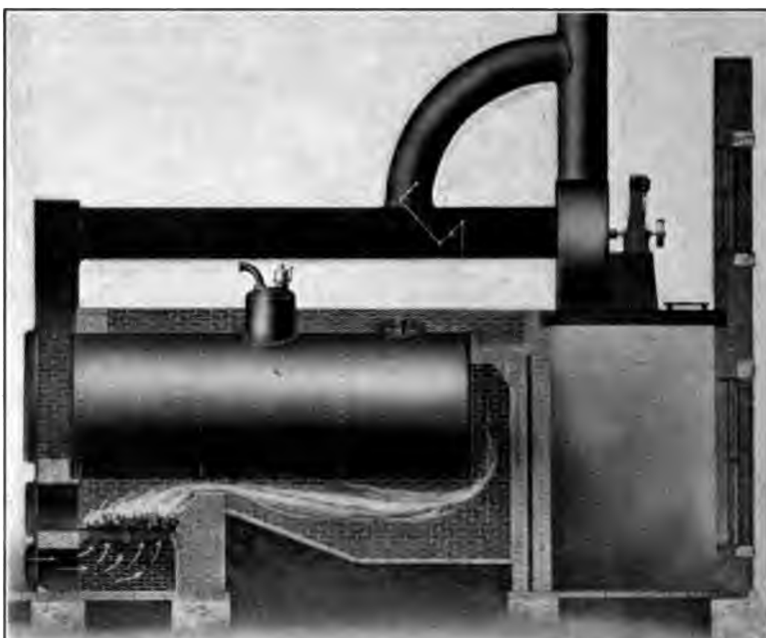


FIG. 115.—Induced draft.

draft and chimney, has made the use of forced draft quite general. In cases where the plant is located at a distance from habitations the induced-draft system may be the best to install.

The best results may be obtained where forced draft is used to force the air through the fire-bed keeping a pressure of 0.01 to 0.03 in. of water in the furnace above the fuel and allowing the chimney draft to carry away the products of combustion. A number of systems using both forced- and induced-draft fans connected by automatic devices for maintaining such a condition have been devised but the older method with hand regulation will usually give better results.

The initial cost of a brick chimney will usually be two or three times that of the mechanical-draft apparatus, but the larger the plant the less will be the relative cost. In small plants, 100 to 150 hp., the cost of a guyed steel stack 75 ft. in height, would be considerable less than that of a mechanical-draft system, and once erected would cost practically nothing for operation, while the power required to operate a fan in a plant of that size would be 5 per cent. or over of the total steam capacity. A tall self-supporting chimney for larger plants, however, is very costly as compared with a fan system of equal capacity. For example, a brick chimney 175 ft. high and 10 ft. in internal diameter capable of furnishing the necessary draft for a 3,000-hp. plant, will cost, including foundation, about $10,000. A duplicate-fan induced-draft system of equivalent capacity will cost about $5,000, a single-fan induced-draft system, $3,500, and a forced-draft system, $2,500.

Capacity of Fans and Power Required.—Ordinary fans are built with radial blades and are usually sized by the height of the casing in inches: thus a 60-in. fan has an impeller of say 42 in. in diameter but the casing height is 60 in. They may be built with a single- or double-inlet. The double inlet impeller is usually considerably wider than the single-inlet. The usual proportions of the runner and case are determined from the "blast area" which is the area through which the fan will discharge giving a velocity equal to the peripheral velocity of the impeller. This, for the standard steel-plate fans, is $a = \frac{WD}{3}$ where a = blast area, W = width of blades at the tips and D = the diameter of the impeller. W is usually about $0.4D$. The radial depth of the blades is usually 0.15 D. The inlet is then about $0.56D$ in diameter and the width of the casing is the same as the diameter of the inlet. Let Q = cubic feet of air per second and let N = r.p.m. Then $a = \frac{0.4D^2}{3} = 0.133D^2$ and the peripheral velocity $= \pi \frac{DN}{60} = \frac{DN}{19.1}$ and $0.133D^2 = \frac{19.1Q}{DN}$ or $D^3 = \frac{144Q}{N}$ for dimensions in feet or roughly for Q = cubic feet per minute, $0.4ND^3 = Q$.

For multivane fans the formula is $1.09ND^3 = Q$. These deliveries are based on certain total pressures (static + velocity) which are dependent on the orifice, the peripheral velocity and type of fan and may be shown by characteristic curves or taken from the tables.

When the conditions are known the theoretical horsepower may be calculated by hp. $= \frac{h_tQ}{6,345}$, where Q = cubic feet per minute and h_t = total head in inches of water. The theoretical horsepower must be divided by the efficiency to give the actual horsepower.

STEEL-PLATE FAN

TABLE OF AIR PRESSURES, CAPACITY AND HORSEPOWER

Water gage, inches	Capacity, cubic feet per minute per square inch of blast area	Theoretical horsepower to move the given volume
0.2	12.2	0.0004
0.4	17.2	0.0011
0.6	21.15	0.0020
0.8	25.0	0.0031
1.0	27.3	0.0043
1.5	33.8	0.0079
2.0	38.8	0.0122
2.5	43.3	0.0169
3.0	47.5	0.0224
3.5	51.4	0.0282
4.0	54.8	0.0344
5.0	61.2	0.0481
6.0	66.9	0.0630

There are many designs of fans on the market which do not agree with these formulas and most builders publish tables for distribution giving sizes, dimensions and performances of their fans under all ordinary conditions. It should be remembered that these published figures refer to the delivery and pressure at the fan outlet. Where delivery is through ducts, the friction and other losses of the ducts must be calculated and added to the conditions at the fan in order that a proper selection may be made.

Multiblade fans of the Sirrocco and Sturtevant types differ from the ordinary fans in having very narrow blades curved forward in the direction of rotation. These blades are from $0.05D$ to $0.1D$ in radial depth and are considerably more efficient than the radial-bladed fans for many services.

High-pressure blowers used for cupola blowing and other high-pressure work have generally cast housings and are made with slightly curved vanes. Their efficiency is also high as compared with the steel-plate fan. Propeller fans of many types are manufactured ranging from the Blackman with very low volumetric efficiency to the Seymour, and McEwen type with 30 to 60 per cent. volumetric efficiency. Guided propeller fans of the Rateau or Parsons type have higher efficiencies.

The characteristic curves for steel-plate, multiblade and cupola fans for the same diameter of impeller and r.p.m. are given below.

When a fan has been tested it becomes possible to draw a characteristic for that fan which will give a view of its performance over a wide range. Such a characteristic is given in Fig. 117.

Stock fans are usually purchased without guarantee but where good results are desired it is better to specify exact conditions and obtain a

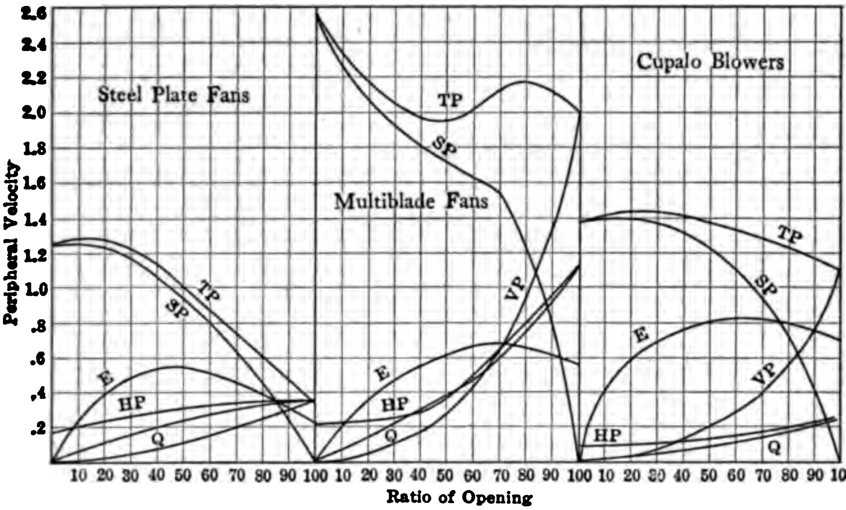


FIG. 116.—Characteristic curves of fans.

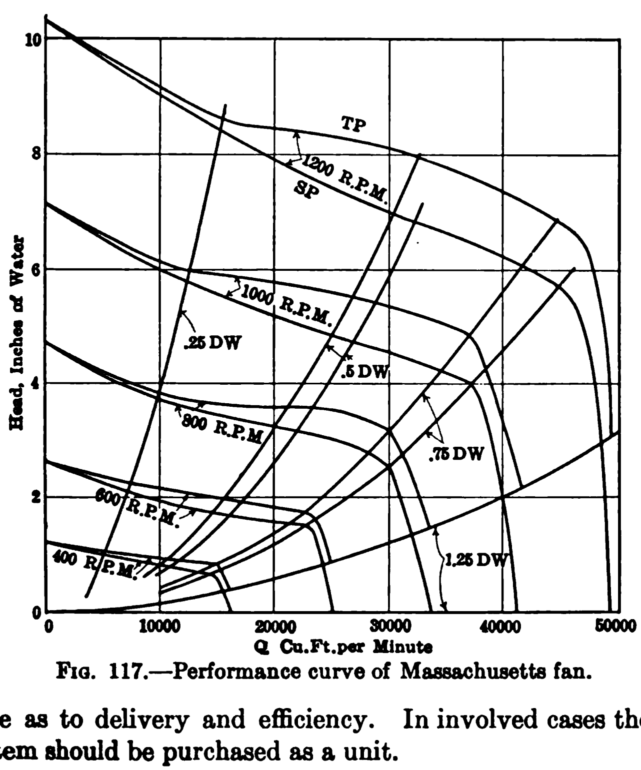


FIG. 117.—Performance curve of Massachusetts fan.

guarantee as to delivery and efficiency. In involved cases the fan and duct system should be purchased as a unit.

Depreciation and Maintenance of Stacks and Mechanical Draft Systems.—The depreciation on a well-designed masonry or concrete stack is very low. A properly constructed steel stack, lined with brick, requires only painting on the outside every 2 years. The depreciation

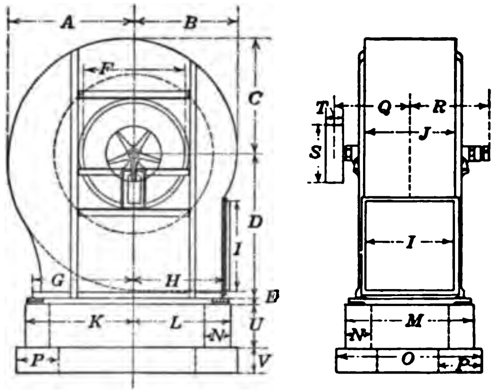


FIG. 118.—Steel-plate fan casing, American Blower Co.

DIMENSIONS OF STEEL-PLATE BLOWERS

Size	A	B	C	D	E	F	G	H	I	J	K
50	24½	19¼	22	29	3	20	21	16¾	18	18	24¼
60	29¼	23¼	26½	34	3	23	24½	19¾	21½	21½	28
70	33½	27	30½	40	3	26	28	23¾	24½	24½	31
80	38½	31	34¾	45	3	30	31½	26½	27	27	35
90	42¾	34¾	39¼	50	3	34	36	30¼	30½	30½	39
100	47½	38½	43¼	55	3	38	39	33¾	34½	34½	42
110	52	42½	47¼	60	3	42	42½	37¼	37½	37½	45½
120	57	46½	51½	66	3	46	47	41¼	41½	41½	51½
140	66	54¼	60	76	3	53	53½	48	48	48	58
160	75¼	62	68½	86	3	60	60½	54	54	54	65
180	84½	69½	77	96	3	68	68½	60½	60	60	72½
200	93¾	77½	85½	106	3	76	75½	66½	66	66	79½
220	103	85	94	116	3	84	82	73	72	72	90
240	112¼	92¾	102½	126	3	92	89	79¼	78	78	97
260	121½	100½	111	136	4	100	96	85	84	84	104
280	130¾	108¼	119½	146	4	108	102	92	90	90	110
300	140	116	128	156	4	116	109¼	98	96	96	117

Size	L	M	N	O	P	Q	R	S	T	U	V
50	20	31	9	39	17	18½	17½	12	4	15	9
60	23	35	9	43	17	21	20	14	4	15	9
70	27	38	9	46	17	23½	22¼	16	4	15	9
80	29½	41	9	49	17	24½	23¾	18	4	15	9
90	33	45	9	53	17	26½	25½	20	4	15	9
100	36½	49	9	57	17	30½	28¾	22	6	15	9
110	40	52	9	60	17	32	30¼	24	6	15	9
120	44	61	9	70	17	35½	33¾	26	6	23	9
140	51	67	13	76	22	39½	37½	30	8	23	13
160	57	73	13	82	22	43	41¼	32	8	23	13
180	63½	80	13	89	22	46	44¼	36	8	23	13
200	69½	86	13	95	22	52½	50	40	10	23	13
220	81	100	13	109	22	59	55½	44	12	35	13
240	87	106	13	115	22	62	58½	48	12	35	13
260	93	112	13	121	22	67	63¼	52	14	35	13
280	100	118	13	127	22	70	66¼	56	14	35	13
300	106	124	13	133	22	74	69¼	60	16	35	13

Speeds, Capacities and Horsepowers of "ABC" Steel-plate Fans at Varying Revolutions

Fan		5	60	70	80	90	100	110	120	140	160	180	200	220	240
	Per V.	785	942	1100	1257	1414	1571	1728	1885	2200	2513	2837	3141	3455	3769
)	Pres. oz.	.017	.025	.034	.044	.055	.068	.082	.100	.134	.175	.231	.273	.335	.401
	Cu. ft.	682	1121	1870	2652	3840	5475	6395	9565	14916	21750	30221	41608	55201	71941
	Hp.	.150	.222	.370	.476	.672	1.01	1.37	2.03	3.46	5.47	7.7	12.0	17.1	25.1
	Per V.	981	1178	1375	1571	1768	1964	2160	2356	2750	3141	3533	3926	4318	4711
5	Pres. oz.	.027	.089	.053	.060	.089	.108	.132	.153	.212	.276	.350	.435	.525	.626
	Cu. ft.	852	1402	2338	3158	4809	6844	7992	11945	18645	27170	37767	52010	68997	99910
	Hp.	.175	.284	.439	.588	.934	1.34	2.06	2.90	5.00	8.15	12.5	19.3	29.2	43.5
	Per V.	1177	1413	1650	1886	2121	2356	2592	2827	3300	3770	4240	4711	5182	5653
50	Pres. oz.	.039	.056	.075	.100	.130	.160	.190	.230	.300	.400	.503	.626	.758	.904
	Cu. ft.	1023	1681	2805	3979	5760	8110	9580	14360	22374	32610	45325	62412	82811	108120
	Hp.	.200	.325	.531	.756	1.27	1.86	2.74	3.90	7.22	11.3	19.6	32.1	46.2	68.6
	Per V.	1374	1649	1925	2200	2474	2749	3024	3297	3850	4380	4947	5496	6046	6596
	Pres. oz.	.053	.076	.104	.134	.172	.212	.258	.306	.420	.554	.687	.848	1.02	1.21
75	Cu. ft.	1194	1962	3274	4622	6729	9594	11200	16715	26100	38043	52883	72814	96626	126089
	Hp.	.225	.393	.647	1.01	1.74	2.46	3.55	5.52	9.91	17.3	27.9	44.2	67.1	103.0
	Per V.	1570	1884	2200	2511	2828	3142	3456	3770	4400	5026	5654	6282	6910	7538
)0	Pres. oz.	.069	.101	.134	.175	.225	.274	.333	.392	.537	.700	.903	1.12	1.34	1.59
	Cu. ft.	1364	2242	3740	5304	7690	10960	12830	19150	29850	43520	60442	83231	110422	143902
	Hp.	.262	.478	.855	1.26	2.05	3.16	4.69	7.01	13.3	23.7	39.2	62.1	96.8	154.5
	Per V.	1766	2120	2475	2829	3182	3534	3888	4241	4950	5654	6360	7065	7774	
25	Pres. oz.	.087	.126	.172	.225	.285	.351	.421	.507	.690	.901	1.14	1.41	1.69	
	Cu. ft.	1534	2523	4207	5969	8655	12334	14385	21500	33560	48680	68000	93634	124217	
	Hp.	.300	.581	1.03	1.57	2.61	4.09	5.95	9.29	17.0	31.1	52.8	87.9	142.5	
	Per V.	1963	2355	2750	3143	3535	3927	4320	4712	5500	6283	7067	7852		
0	Pres. oz.	.109	.056	.213	.280	.360	.430	.520	.630	.860	1.12	1.48	1.73		
	Cu. ft.	1706	2793	4675	6332	9600	13705	16000	23950	37310	54200	75558	104036		
	Hp.	.375	.684	1.22	1.79	3.32	4.97	7.44	11.6	22.5	41.2	71.7	121.4		
	Per V.	2159	2591	3025	3457	3889	4319	4731	5183	6050	6911	7774			
5	Pres. oz.	.131	.189	.258	.337	.426	.526	.623	.756	1.04	1.35	1.71			
	Cu. ft.	1876	3083	5142	7294	10578	15773	17394	26278	41020	58328	83104			
	Hp.	.436	.821	1.45	2.35	3.92	6.09	9.09	14.5	29.4	54.7	89.3			
	Per V.	2355	2826	3300	3771	4242	4712	5184	5654	6600	7539				
	Pres. oz.	.160	.225	.302	.401	.520	.630	.760	. 910	1.26	1.62				
	Cu. ft.	2046	3363	5610	7957	11520	16250	19200	28800	44750	63629				
	Hp.	.500	.975	1.73	2.86	4.63	7.44	11.4	18.1	37.5	69.3				
	Per V.	2747	3297	3850	4399	4949	5447	6048	6597	7700					
	Pres. oz.	.216	.306	.418	.550	.693	.850	.970	1.25	1.68					
	Cu. ft.	2387	3923	6545	9282	13410	19110	22395	33400	52206					
	Hp.	.663	1.28	2.38	3.89	6.65	10.7	17.2	28.3	55.8					
	Per. V.	3140	3768	4400	5028	5656	6282	6912	7540						
	Pres. oz.	.277	.399	.546	.713	.904	1.14	1.42	1.63						
	Cu. ft.	2729	4384	7480	10620	15400	21950	25574	38300						
	Hp.	.750	1.70	3.19	5.04	9.34	15.3	25.2	39.2						

ıd maintenance charges on a mechanical-draft system will range from to 15 per cent. of the cost and in the case of induced systems may considerably higher.

Efficiency With Stacks and Mechanical Draft Systems.—With ıechanical draft a much thicker fire can be maintained on the grates, hus permitting a high rate of combustion and minimum draft per pound of fuel, both of which result in increased boiler efficiency. Where the forced-draft system is used a proper manipulation of stack dampers and fan speeds leads to a balanced draft from which exceedingly good results can be obtained. In this case the chimney has only to furnish

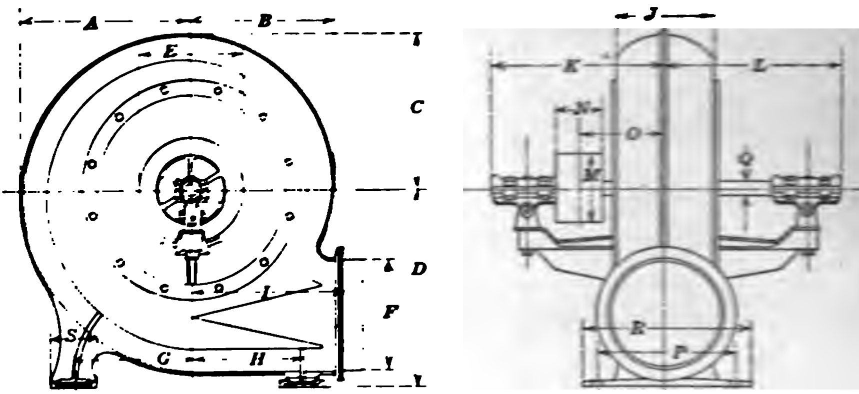


Fig. 119.—Cast-iron case volume blower. American Blower Co.

Dimensions

Size	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S
1	7⅝	6⅜	7	9	4¾	5	5¼	4¾	6⅜	4	8¼	8¼	3	2½	3⅜	6	1 1/16	7	2
2	9⅛	7⅛	5½	10¾	5⅝	6	6¼	5½	8¼	5	11 5/16	11 5/16	4	3	5½	6¾	1 3/16	8½	2½
3	10⅝	8⅞	9¾	12½	6⅝	7	7½	7	9¾	5¾	12¾	12¾	5	3½	5½	8	1 5/16	10	3
4	13⅜	11½	12⅝	16	8 5/16	9	9½	8¾	12 5/16	7½	15½	15½	6	4	7	11½	1 1/16	13	3¾
5	16¾	14⅛	15⅜	19¾	10⅜	11	11¾	10¾	15⅛	9	17½	17½	7	4½	8	13	1 3/16	16	4½
6	20⅛	16⅞	18 7/16	23½	12½	13	14	13	17⅞	10¾	20	20	8	5½	9½	15½	1 5/16	19	5¼
7	23⅛	19⅜	21¼	27	14½	15	16	15⅛	20⅝	12½	23⅛	23⅛	9	6½	11⅝	18	1 7/16	22	6
8	26	21¾	24	30⅝	16½	17	18⅛	17¼	23¼	14¼	26	26	10	7½	13	20½	1 9/16	25	6¾
9	29	24⅛	26¾	34¼	18⅜	19	20¼	19½	26	16	29	29	11	8½	14½	23	1 11/16	28	7½

Speed, Capacity and Horsepower Required for the "ABC" Volume Blowers

Size	Dia. of wheel	Width at periphery	Cir. in feet	1½ Oz. pressure R.p.m.	1½ Oz. pressure Cu. ft.	1½ Oz. pressure Hp. net	1¾ Oz. pressure R.p.m.	1¾ Oz. pressure Cu. ft.	1¾ Oz. pressure Hp. net	2 Oz. pressure R.p.m.	2 Oz. pressure Cu. ft.	2 Oz. pressure Hp. net
1	8½	2	2.22	3300	348	0.350	3560	376	0.445	3810	402	0.545
2	10½	2¾	2.75	2650	512	0.520	2880	554	0.655	3080	590	0.800
3	12	3¼	3.15	2320	711	0.728	2510	770	0.918	2690	822	1.17
4	15½	4⅜	4.06	1800	1210	1.24	1950	1305	1.55	2080	1395	2.53
5	19	5½	4.98	1470	1830	1.87	1590	1980	2.35	1700	2110	3.83
6	22½	6½	5.90	1240	2600	2.66	1340	2810	3.33	1440	3000	5.45
7	26	7½	6.80	1075	3420	3.50	1160	3700	4.37	1250	3960	7.20
8	29½	8½	7.73	950	4130	4.54	1025	4800	5.68	1100	5125	9.30
9	33	9½	8.65	845	5580	5.72	915	6020	7.15	980	6440	11.7

Size	Dia. of wheel	Width at periphery	Cir. in feet	3 Oz. pressure R.p.m.	3 Oz. pressure Cu. ft.	3 Oz. pressure Hp. net	4 Oz. pressure R.p.m.	4 Oz. pressure Cu. ft.	4 Oz. pressure Hp. net	5 Oz. pressure R.p.m.	5 Oz. pressure Cu. ft.	5 Oz. pressure Hp. net
1	8½	2	2.22	4670	492	1.00	5400	568	1.54	6100	635	2.15
2	10½	2¾	2.75	3770	725	1.47	4360	835	2.26	4900	935	3.17
3	12	3¼	3.15	3300	1005	2.06	3810	1160	3.17	4300	1300	4.44
4	15½	4⅜	4.06	2560	1705	3.46	2960	1970	5.35	3330	2200	7.45
5	19	5½	4.98	2080	2590	5.75	2410	2980	8.07	2720	3340	11.3
6	22½	6½	5.90	1760	3670	7.45	2040	4250	11.5	2300	4750	16.2
7	26	7½	6.80	1530	4850	9.85	1770	5600	15.2	2000	6250	21.2
8	29½	8½	7.73	1340	6270	12.8	1550	7250	19.7	1750	8100	27.4
9	33	9½	8.65	1200	7875	16.0	1390	9100	24.7	1570	10180	34.4

sufficient draft to remove the products of combustion, while the forced draft maintains the combustion at the proper rate and produces a slight plenum above the fire, thus preventing the large losses due to inrush of cold air when the fire-doors are opened. With mechanical draft the amounts of air and pressure can be readily regulated for any sudden increase or decrease of the requirements practically independent of boiler performance. Damp muggy days appreciably affect the draft of the chimney as do adverse air currents and high winds, but these

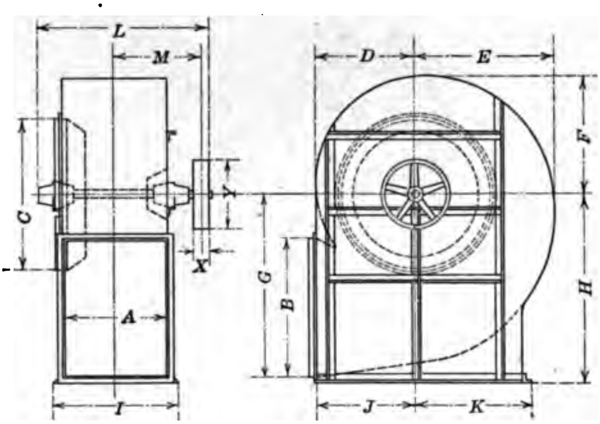


Fig. 120.—Niagara conoidal fan.

Dimensions of Niagara Conoidal Fan

Overhung Pulley Full Housing—Bottom Horizontal Discharge

Dimensions in Inches

Size	A	B	C	D	E	F	G	H	I	J	K	L	M	X	Y
3	12	15 3/4	17 1/4	11 3/16	15 7/8	13 1/4	20 13/16	22	16 1/4	11 3/4	14	27 1/2	15	3 1/2	8
3 1/2	14	18 3/8	20	13	18 9/16	15 7/16	24 1/4	25 1/2	18 1/4	13	16	29 1/2	16	3 1/2	9
4	16	21	22 3/4	14 7/8	21 3/16	17 5/8	27 3/4	29	20 1/4	14 7/8	18	31 1/2	17	3 1/2	10
4 1/2	18	23 5/8	25 3/4	16 3/4	23 7/8	19 7/8	31 1/4	32 1/2	22 1/4	16 3/4	20	33 1/2	18	3 1/2	11
5	20	26 1/4	28 1/2	18 5/8	26 1/2	22 1/16	34 11/16	36	24 1/4	18 5/8	22	36	19 1/2	3 1/2	12
5 1/2	22	28 7/8	31 1/2	20 7/16	29 1/8	24 1/4	38 3/16	39 1/2	26 1/4	20 7/16	24	37	19 1/2	3 1/2	14
6	24	31 1/2	34 1/4	22 5/16	31 13/16	26 1/2	41 5/8	43	28 1/4	22 5/16	26	41 3/4	22	4 1/2	16
7	28	36 3/4	39 3/4	26	37 1/8	30 7/8	48 9/16	50 3/8	32 1/4	26	30	50	25 1/2	5 1/2	18
8	32	42	45 1/2	29 3/4	42 3/8	35 5/16	55 1/2	56 3/4	36 1/4	29 3/4	34	56	29	6 1/2	20
9	36	47 1/4	51 1/4	33 1/2	47 11/16	39 3/4	62 7/16	64	40 1/4	33 1/2	38	63 1/2	32	8 1/2	24
10	40	52 1/2	56 3/4	37 3/16	53	44 1/8	69 3/8	70 3/4	44 1/4	37 3/16	42	67 1/2	34	8 1/2	26
11	44	57 3/4	62 1/2	40 15/16	58 5/16	48 1/2	76 5/16	78	49 1/4	40 15/16	46 1/2	75 1/2	38	8 1/2	28
12	48	63	68	44 5/8	63 5/8	52 15/16	83 1/4	85	53 1/4	44 5/8	50 1/2	81	41	10	30
13	52	68 1/4	73 1/2	48 3/8	68 7/8	57 3/8	90 3/16	92	58 1/4	48 3/8	55	85 5/8	43	11	34
14	56	73 1/2	79	52 1/16	74 3/16	61 3/4	97 1/8	99	62 1/4	52 1/16	59	95 1/2	48	13	36
15	60	78 3/4	83 3/4	55 3/4	79 1/2	66 3/16	104 1/16	106	66 1/4	55 3/4	63	100 1/2	50	15	38
16	64	84	90 1/4	59 1/2	84 3/4	70 5/8	111	112 1/2	71 1/4	59 1/2	67 1/2	109	54		40
17	68	89 1/4	96	63 1/4	90 1/16	75	117 15/16	119 1/2	76 1/4	63 1/4	72	115	56 1/2		44
18	72	94 1/2	101 1/2	66 15/16	95 3/8	79 7/16	124 7/8	126 1/2	80 1/4	66 15/16	76	122 1/2	61		46
19	76	99 3/4	107	70 11/16	100 11/16	83 13/16	131 13/16	133 1/2	84 1/4	70 11/16	80	128	63		48
20	80	105	112 3/4	74 3/8	106	88 1/4	138 3/4	140 1/2	88 1/4	74 3/8	84	130	63 1/2		50

conditions with mechanical draft will not affect the burning of coal, since the amount of chimney draft used is much smaller than the capacity of the chimney.

It is claimed that smokeless combustion is more readily effected with artificial draft than with natural draft, as a thicker fire can be carried and the correct proportion of air more readily adjusted.

FIG. 121.—Runner of conoidal fan, Buffalo Forge Co.

CAPACITY TABLE BUFFALO FANS

Fan no.	Mean dia. of blast-wheel	Area of outlet, sq. ft.	½" Total press. or 0.288 oz.			1" Total press. or 0.577 oz.			2" Total press. or 1.154 oz.			3" Total press. or 1.734 oz.		
			Rev.	Vol.	Hp.	Rev.	Vol.	Hp.	Rev.	Vol.	Hp.	Rev.	Vol.	Hp.
3	15¾"	1.31	478	1,720	0.19	675	2,440	0.54	955	3,450	1.54	1,169	4,220	2.83
3½	18¼"	1.79	409	2,350	0.26	579	3,320	0.74	818	4,690	2.09	1,002	5,750	3.85
4	21"	2.33	358	3,070	0.34	506	4,340	0.97	716	6,130	2.73	877	7,510	5.02
4½	23½"	2.95	318	3,880	0.43	450	5,490	1.22	636	7,760	3.46	780	9,500	6.30
5	26"	3.64	287	4,790	0.53	405	6,770	1.51	573	9,580	4.27	702	11,730	7.84
5½	28¾"	4.41	260	5,800	0.65	368	8,200	1.83	521	11,590	5.47	638	14,190	9.49
6	31½"	5.25	239	6,900	0.77	338	9,750	2.17	477	13,790	6.15	585	16,890	11.30
7	36½"	7.14	205	9,400	1.05	289	13,280	2.96	409	18,770	8.37	501	23,000	15.40
8	42"	9.33	179	12,260	1.37	253	17,340	3.87	358	24,520	10.90	439	30,040	20.10
9	48"	11.81	159	15,520	1.73	225	21,950	4.89	318	31,020	13.80	390	38,010	25.40
10	52"	14.58	143	19,160	2.14	203	27,090	6.04	286	38,310	17.10	351	46,930	31.40
11	57½"	17.64	130	23,180	2.58	184	32,780	7.31	260	43,360	20.70	319	56,780	38.00
12	63"	21.00	119	27,590	3.08	169	39,010	8.70	239	55,170	24.60	292	67,570	45.20
13	68"	24.65	110	32,370	3.61	156	45,780	10.20	220	64,730	28.90	270	79,300	53.00
14	73"	28.56	102	37,550	4.19	145	53,100	11.80	205	75,090	33.50	251	91,970	61.50
15	78¼"	32.81	96	43,100	4.80	135	60,960	13.60	191	86,200	38.40	234	105,580	70.60
16	83½"	37.33	90	49,040	5.47	127	69,300	15.50	179	98,060	43.70	219	120,130	80.30
17	89"	42.14	84	55,370	6.17	119	78,300	17.50	189	110,720	49.40	206	135,620	90.70
18	94"	47.25	80	62,060	6.92	113	87,780	19.60	159	124,110	55.30	195	152,020	101.70
19	99"	52.65	75	69,160	7.71	107	97,800	21.80	151	138,280	61.70	185	169,400	113.30
20	105"	58.33	72	76,040	8.54	101	108,370	24.20	143	153,250	68.30	175	187,680	125.50

The advantages of forced and induced draft may be summed up as ›ws: (1) Draft not limited by atmospheric conditions or height of k; (2) increased capacity of plant, within limits, at will; (3) possibility urning inferior fuel with advantage.

Disadvantages: (1) High operating cost of the machine; (2) occupies :e which often is valuable; (3) uses from 1 to 5 per cent. of the steam :rated by the boiler.

PROBLEMS

2. The height of a chimney at the plant of Fall River Iron Co., Boston, is 350 ft.; nal diameter, 11 ft. Determine the number of pounds of coal that can be burned iour and the horsepower of the chimney.

3. The power plant of the Passaic Print Works, Passaic, N. J., has a chimney 9 ft. ternal diameter which handles the gases from 13,855 lb. of coal per hour. Deter- : the height of the stack.

4. The plant of the Amoskeag Mills, Manchester, N. H., has a stack 230 ft. high h handles the gases from 19,195 lb. of coal per hour. What is the diameter?

5. (*a*) Two boilers in the market have the following heating surface:

(*A*) Is a water-tube boiler with 3,350 sq. ft.
(*B*) Is a fire-tube boiler with 2,590 sq. ft.

Jnder test (*A*) actually evaporated 144,000 lb. of water in 12 hr., and (*B*) 108,000 What was the approximate per cent. of rating carried by each boiler?

b) On the basis of the evaporation given in (*a*), and usual operating conditions, much coal was fired under each boiler per 12 hr. if the coal contained 10 per cent. sture?

c) On the basis of (*b*), what would be the diameter and height of stack required his plant, if one stack serves both boilers?

d) With a stack of the dimensions determined in (*c*), how does its commercial epower, as given in the table on page 176, compare with that of the given plant?

16. A plant containing five 300-hp. boilers, four of which are in continuous service er approximately 33 per cent. overload, has a chimney 7½ ft. in internal diameter 160 ft. high. Additional steam demands will require the installation of two 300- boilers, these to run at about the same overload. The fuel used is run of mine minous coal, about 13,000 B.t.u. per pound. Will the present stack be of sufficient acity?

17. In connection with the above plant, it is finally decided to erect another stack iave sufficient capacity to handle three 300-hp. boilers at 50 per cent. overload, ning the same grade of coal. The boilers have a ratio of heating surface to grate ace of 50 to 1. Estimate the height and the diameter of stack to be used.

18. Determine a "Handy" (approximate) figure for the required capacity of (1) rced-draft fan in terms of cubic feet per minute per pound of coal burned per hour; an induced-draft fan.

19. It is desired to provide a forced-draft fan of the steel-plate type for a battery of lers of 1,000 hp. rated capacity but from which it is intended to be able to take 100 cent. overload. The boilers are to be equipped with underfeed stokers, on which manufacturers guarantee 100 per cent. boiler overload capacity with 3.5 in. of er draft. Determine the impeller diameter and width, the r.p.m., and the horse- rer rating of engine drive required for a steel-plate type of draft fan.

CHAPTER VIII

SMOKE AND SMOKE PREVENTION

Smoke is a result of imperfect or improper combustion. The Standard Dictionary states that smoke is the volatilized products of the combustion of an organic compound, as coal, wood, etc., charged with fine particles of carbon. The definition of the word "smoke" given in the recent (1915) report on Smoke Abatement and Electrification of Railway Terminals in Chicago is "the gaseous and solid products of combustion, visible and invisible, including, in the case of certain industrial fires, mineral and other substances carried into the atmosphere with the products of combustion."

Smoke consists then of:

1. Visible properties.
2. Solid constituents.
3. Gaseous constituents.

The ordinary conception of smoke is based upon the effect of particles of carbon upon the eye, and the fact is generally lost sight of that, other than from an esthetic standpoint, more damage may result from the nearly colorless gases issuing from a chimney than from the more offensive black smoke. So pronounced is this color impression that the majority of ordinances relate specifically to the density of the smoke as determined by color charts. The universal standard is a system of charts known as the Ringelmann system, which, when placed at the proper distance from the observer, give graduations of gray between pure white and black. Although the more harmful portions of smoke are practically colorless, it is nevertheless true that the color graduation may be an index of the efficiency of combustion and may indicate the proportion of injurious gases that are issuing from a given stack.

In the ordinary furnace under a steam boiler we find the grate upon which the fuel is placed; the openings in the grate through which air is supplied; the combustion chamber in which the oxygen of the air and the gases of combustion are thoroughly mixed; and the chimney or stack for producing the necessary draft and for carrying away the products of combustion.

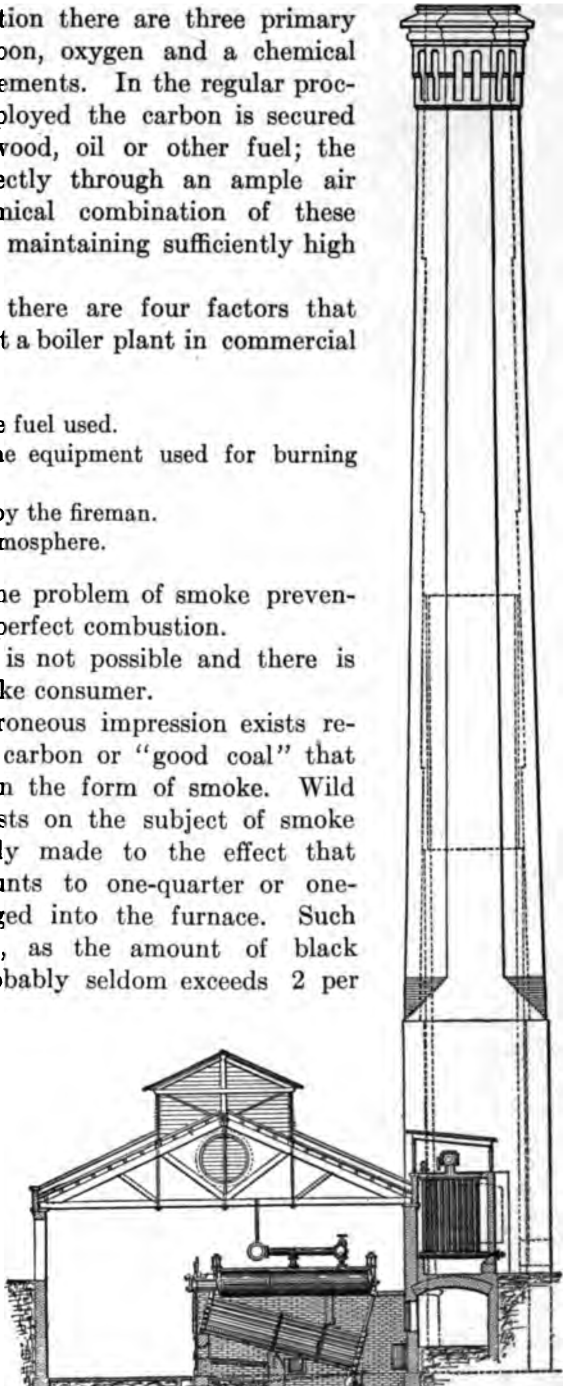
combustion there are three primary ely, carbon, oxygen and a chemical these elements. In the regular procally employed the carbon is secured coal, wood, oil or other fuel; the red directly through an ample air he chemical combination of these ıred by maintaining sufficiently high

ınt that there are four factors that at extent a boiler plant in commercial moke:

ter of the fuel used.
ıter of the equipment used for burning

ɛercised by the fireman.
ɔf the atmosphere.

ɜ that the problem of smoke prevenɔlem of perfect combustion.
ımption is not possible and there is s a smoke consumer.
ımon erroneous impression exists reɔunt of carbon or "good coal" that mneys in the form of smoke. Wild ɛnthusiasts on the subject of smoke requently made to the effect that ɜs amounts to one-quarter or oneel charged into the furnace. Such absurd, as the amount of black ıing probably seldom exceeds 2 per ıt.
de up , tar, and mall senic. from the n inıestic from



ate, air passage, combustion chamber and stack of typical plan

	Industrial plants, per cent.	Domestic plants	
		Kitchen, per cent.	Drawing room, per cent.
Carbon	27.00	53.34	37.22
Hydrogen	1.68	3.68	3.51
Tar	1.14	12.46	40.38
Ash	61.80	17.80	4.94

It is readily seen that the amount of carbon and tar from domestic chimneys is relatively far greater than from industrial chimneys. It is also correspondingly apparent that the ash from domestic chimneys is relatively far less than from industrial chimneys. Domestic chimneys are seen, therefore, to have a record that is far from clear and must be taken into account in considering the ultimate solution of the smoke problem.

Effects of Smoke.—Briefly, the effects of smoke may be summarized as:

1. Effect on buildings and building materials.
2. Effect on vegetation.
3. Effect on weather conditions.
4. Effect on health and conduct.

The smoke nuisance has become such an important factor in connection with urban power-plant installations that it deserves the serious consideration of all students of engineering. From the exhaustive report of the Chicago Association of Commerce relating to Smoke Abatement and Electrification of Railway Terminals (1915) the following conclusions are taken:

"A survey of the atmosphere of several of the world's great cities shows an improvement in atmospheric conditions during recent years. It shows also that Chicago suffers less from the effects of smoke than certain other large cities of this and other countries.

The comparison of the air of cities with that of the country has revealed characteristics which may and apparently must be attributed to the smoke of the cities; but it has also been shown that they may in part be attributed to other sources, as, for example, leakage from gas mains, the pollution due to sewers, the dust of the streets and decaying organisms. Air analysts have admittedly not been able to separate the products of combustion as dispersed in the air from other agents of air pollution.

The industrial activity of all important cities has brought about an increase in coal consumption which is greater than the increase in population. Smoke formation and the consequent pollution of the atmosphere by smoke have in recent years tended to increase, and have done so except so far as the adoption of various means in smoke prevention have proved effective. The fact is repeatedly pointed out that, in securing results of scientific value for use in abating smoke, one individual and no one city can accomplish the work that must be done. e observations must be numerous and must extend over decades.

The fact appears firmly established that there is a well-defined relation be-een smoke and fog and that the presence of smoke induces fog. It is agreed it sunshine is a function of the amount of smoke present in the atmosphere.

Certain investigations have shown that the amount of carbon dioxide in the nosphere of cities is, as a rule, only about 1 per cent. greater than that in coun-r air. The sulphur compounds in the atmosphere are generally due to the mbustion of coal.

Among the sources of pollution of city air by smoke, the world over, domestic imneys are conspicuous. The mention of them by observers and students is uch more frequent than the mention of any other source.

The most successful means which have been employed to abate smoke have cluded not only legal prohibition but also the development of coöperative and lucative measures.

With reference to the effects of smoke, the following conclusions seem istified by the literature on the subject:

(a) There is a general agreement among sanitary authorities that polluted r is harmful to health, but at the present time there exists no accurate method measuring this harm nor of determining the relative responsibility of the differ-it elements which enter into the mixture of gases and solids commonly referred as atmospheric air.

(b) The direct effects of smoke or of any of its attributes, including soot, ust and gases, in amounts which may ordinarily pervade the atmosphere of a noky city are not shown to be detrimental to persons in normal health, but the neral physical tone is lowered as the result of long-continued breathing of pol-ited air.

(c) The direct effect of smoke upon those who are ill has been most extensively udied in connection with tuberculosis and pneumonia. It appears that smoke oes not in any way stimulate the onset of the tubercular process nor militate gainst the rapidity of recovery when once this disease has been contracted, but at it has a direct antiseptic effect and tends to localize the disorder. In cases f pneumonia, the effect becomes seriously detrimental.

(d) The tarry matter and sulphur compounds present in coal smoke have een shown by experiments to affect certain classes of vegetation when applied n sufficient quantities.

(e) Smoke is popularly regarded as a source of loss and damage in its effects pon building materials, objects of virtu, clothing and other property. While hese effects of smoke seem obvious, it has not been possible to estimate their xtent with any degree of accuracy."

"Smokeless combustion of bituminous coal, as defined by present practice volves compliance with certain well-defined principles, the more important of which may be described as follows:

1. The fresh coal should be introduced into the furnace at such a point and listributed in such manner that the gases distilled from it will be required to pass over the incandescent portions of the fire. Observance of this condition exposes

the distillates to high temperatures, aids in their ignition and thereby promotes their combustion. The distillates, if not thoroughly burned are prolific sources of smoke.

2. The stream of gases arising from the fresh fuel must be heated as quickly as practicable and must be kept at a high temperature until the process of combustion is well advanced. The presence of a firebrick arch under which the distillates may be burned is an aid in securing this condition.

3. The interposition of heat absorbing surfaces in close proximity to the fresh coal or the burning distillates tends to cool the gases, to suppress combustion and to produce smoke.

4. The admission of air, by which combustion is stimulated, should be provided for at proper points and should be subject to careful regulation.

5. The proportions of the furnace should be such as will provide an ample flameway. This condition is necessary in order that the time occupied by the gases in passing through the furnace may be sufficient to permit them to burn completely. Where the length of the furnace is limited, the flameway may be extended by the use of baffle arches which require the gaseous stream to meander through the furnace, producing in effect an elongation of the flameway and promoting the mixing of the gases. A brick arch in the comparatively small furnace of the locomotive serves to increase the length of the flameway, promotes the intermixing of gases and maintains the temperature required for igniting the gases.

6. Where the dimensions of the furnace are necessarily restricted, and where the air admitted cannot be perfectly distributed, the use of small steam jets with induced air discharged into the furnace serves to promote the mixture of gases, and by so doing, to improve combustion. The use of such jets with induced air on locomotive fireboxes is known to be of material service in suppressing visible smoke."

To obtain entirely satisfactory results from hand-fired furnaces, certain recommendations for the guidance of firemen are laid down by different authorities. Among these are the following:

1. Fuel should be supplied to the fire periodically in small quantities. "A furnace well designed and operated will burn many coals without smoke up to a certain number of pounds per hour, the rate varying with different coals depending on their chemical composition. If more than this amount is burned, the efficiency will decrease and smoke will be made owing to the lack of capacity to supply air and mix gases."[1]

2. The accepted methods of supplying coal to hand-fired furnaces are four in number, as follows:

(*a*) The "spreading or sprinkling" method; uniform stoking of the entire surface of the grate.

(*b*) The "coking" method; covering the front part of the grate after pushing back the glowing coal.

(*c*) The "ribbon" method; partially covering the surface of the grate by stoking the entire length of the grate and only partially covering the sides, or by stoking one-half of the grate surface.

[1] United States Geological Survey, *Bulletin* 373, 1912.

(*d*) The "alternate" method; used when the grate has two or more doors through which to feed the fuel."

"A review of the literature relating to mechanical, physical and chemical means of abating smoke shows:

1. That among the means which have been suggested to reduce the amount of smoke in the atmosphere of cities are:

(*a*) The removal of fuel consuming industries to points remote from the city.

(*b*) The construction of smoke sewers, or community chimneys, of such size and height as to permit of directing the discharges from many flues into one stack and thereby delivering the combined stream far above the city.

(*c*) The establishment of central heating and power plants combining the activities of many small coal consuming plants into a few large centers which may possibly be located at points removed from areas of congested population.

(*d*) The employment of devices for washing smoke discharges before emission into the atmosphere.

(*e*) The condensation and deposition of smoke particles by means of electric devices.

(*f*) The abolition of many small coal fires through an extension of the use of gas and electricity.

(*g*) Improvement in methods of firing.

2. That fires of bituminous coal may be maintained without becoming sources of visible smoke, providing certain principles are recognized in the design of furnaces and in the manner in which they are fired.

3. That it is possible to secure smokeless combustion of fuel fired under stationary boilers by hand-firing, though such a result implies careful supervision.

4. That many types of automatic stokers are available, the operation of which, under favorable conditions, is unattended by the production of visible smoke.

5. That various aids to combustion are recognized which, when applied to furnaces, tend to suppress smoke. The more important of these are:

(*a*) The brick arch as applied to stationary boilers and as applied to locomotive boilers.

(*b*) The use of baffle walls in furnaces.

(*c*) The use of the steam jet with induced air for accelerating the process of combustion."

The complete elimination of smoke in Chicago is set forth by the Commission as follows:

"The complete electrification of Chicago railway terminals would not suffice to make the city smokeless. It has been estimated that only from 30 to 50 per cent. of all the smoke which pollutes the atmosphere of Chicago comes from locomotives. The remainder is from domestic and industrial fires, small and large. Large industrial fires may, through the use of appliances which are well known, readily be made smokeless, and it is to fires of this class that the City Smoke Department has given most attention. A large percentage of the total smoke comes from domestic fires or from industrial fires so small as to make difficult the bestowal of sufficient care upon them to secure the prevention of smoke. Any plan, therefore, which aims at the development of a smokeless city must deal effectively with such small fires. Interpreting this problem, in terms of Chicago's fuel resources and our present knowledge of the art, requires a very strict enforcement of City ordinances not only for the suppression of smoke from equipment already installed, but very stringent regulations governing installation of new furnace apparatus. With the present administration of the Department of Smoke Prevention, under the direction of a competent commission, it is probable that a new and better ordinance than that now in force could be of limited value at the present time, but the future will impose requirements not dealt with at present.

While the policies and methods at present employed in the conduct of the Department of Smoke Inspection are, without doubt, correct in a general way, there are several matters that should receive particular attention, as follows:

(*a*) Much smoke is produced at night, on Sundays and holidays, when the city smoke inspectors are not at work. As this smoke is fully as objectionable as the smoke made during other times, the immediate establishment of smoke inspection service during these periods is recommended.

(*b*) When an enlarged organization of the Department has been effected, the ordinance should be changed to deal with different grades of smoke instead of dense smoke only, as at present.

(*c*) The small heating boiler used in apartment houses, small flat buildings and residences is a very crude appliance. The state of the art in its application to apparatus of this character is in the same undeveloped stage as that of the standard type of boiler furnace years ago and its development has remained stationary while that of the power boiler has made great advances. Therefore, the development of improved types of low-pressure heating boilers should be encouraged and within reasonable time their use made compulsory.

(*d*) The use of such smokeless fuels as gas and coke should be encouraged. Each is an ideal fuel for domestic and small intermittent fires. The only limitation to their employment is that of cost and anything that will reduce that cost should be encouraged.

(*e*) The extension of the plan for supplying steam for heat and power to adjacent buildings from a plant centrally or conveniently located with reference to those to be served, is a scheme having great possibilities for the elimination of smoke, as it makes possible the generation of steam under more favorable conditions than prevail in small plants.

(*f*) The installation of automatic stokers in the smaller steam-power plants should be enforced.

(*g*) As there appears to be no way of operating river tug boats without objectionable smoke except by the use of anthracite coal, the boats on the Chicago River should be required to use such fuel.

(*h*) Passenger and freight steamers in the Chicago River should either be required to use a better grade of fuel than at present, or mechanical stokers should be installed to use the fuel now employed.

(*i*) There are many special problems in connection with the prevention of smoke that have and will present themselves from time to time, requiring special and particular study. Such problems consist in the suppression of smoke from furnaces of rolling mills, brickyards, malleable-iron plants, terra-cotta works and in similar industries, as well as from automobiles, etc.

CHAPTER IX

BOILER AUXILIARIES

Feed Pumps.—In the majority of cases an extremely wasteful steam engine is used to operate the steam pump for supplying the boiler with feed water. As the power required for pumping the feed water is only a small portion of the entire amount, an extremely uneconomical pump does not represent a great percentage loss of the entire fuel. Further than this, as the exhaust steam from the auxiliaries is usually needed for heating the feed water the actual steam consumption chargeable to the feed pump is not over 10 per cent. of its water rate.

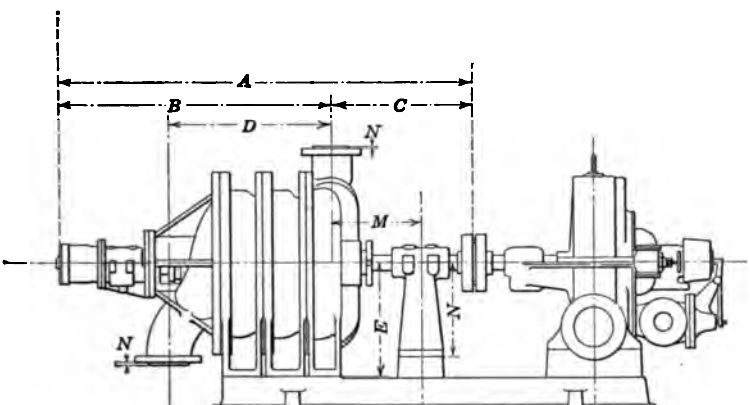


FIG. 123.—Worthington three-stage centrifugal feed pump with Terry turbine drive.

Steam Consumption of Feed Pumps.—With compound steam ends well lagged and covered 100 lb. of steam per indicated horsepower-hour should be safe consumption for feed pumps of large size, while in small pumps, 200 lb. appears to be nearer the mark. The pump efficiency should not be less than 80 per cent.

The following illustrates the method of determining the amount of steam required by the feed pump to supply a given amount of water against a given head.

Suppose the main engine, radiation and leakage, and all auxiliaries with the exception of the feed pump, require 8,000 lb. of steam per hour when operating at full rated load, and the boiler pressure is 150 lb. gage.

The feed pump must now pump not only the 8,000 lb. of water against 150 lb. pressure, but, in addition, the actual amount of steam required to operate the feed pump itself.

Assuming the economy of the pump to be 200 lb. of steam per indicated horsepower-hour and its efficiency to be 80 per cent., and letting "s"

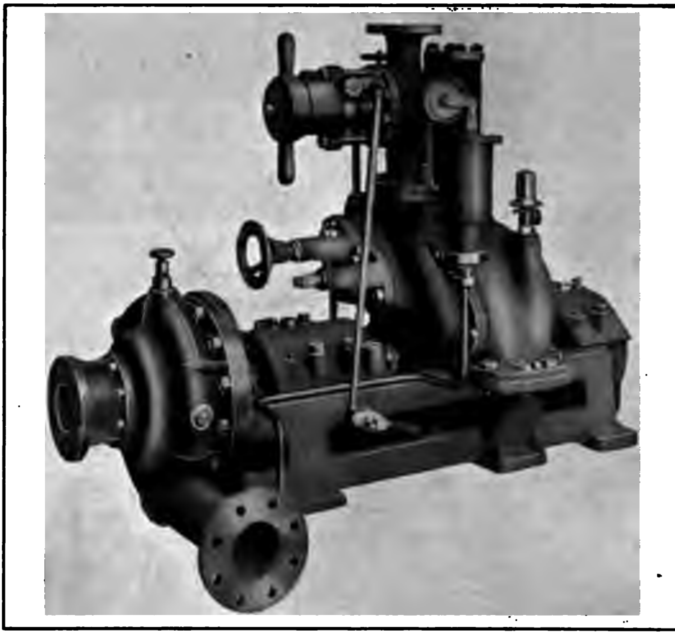


Fig. 124.—Turbine-driven centrifugal feed pump, A.E.G., Berlin.

represent the total steam used by the feed pump per hour, the value of "s" may be found from the following:

$$s = \frac{(8{,}000 + s) \times 150 \times 2.31 \times 200}{3{,}300 \times 60 \times 0.80}$$

Centrifugal Feed Pumps.—About 10 years ago it was found to be a comparatively easy matter to design a centrifugal pump to deliver water at pressures in excess of 200 lb., and the search for an ideal feed pump ended in the adoption of centrifugal pumps, using two to five stages, driven by steam turbines or electric motors. These pumps were continuous in their action, thus putting no severe strains on the feed piping, as did the intermittent action of the old duplex or triplex pumps. If the feed valves were all shut off by some chance, no accident occurred, since the centrifugal pump simply churned the water, but did not deliver it. It was found that the pumps were much smaller, required almost no attendance and a large saving was made, due to the absence of pump-valve renewals, which with hot water, had amounted to a rather large figure. The steam consumption of the turbines, even when run non-condensing, was reasonably low, due to the high rate of rotation, and did not increase with age, as in the case of the reciprocating pumps. It was no uncommon thing to find the steam consumption of a 6 by 4 by 6-in. duplex feed pump to be 180 lb. per horsepower when new, and this consumption would increase with the age of the pump until the pump was dismantled and the steam valves ground tight. Even when the centrifugal pump ran at as low a speed as 1,800 r.p.m., the steam consumption of the turbine, when run non-condensing, would not exceed 50 lb. per

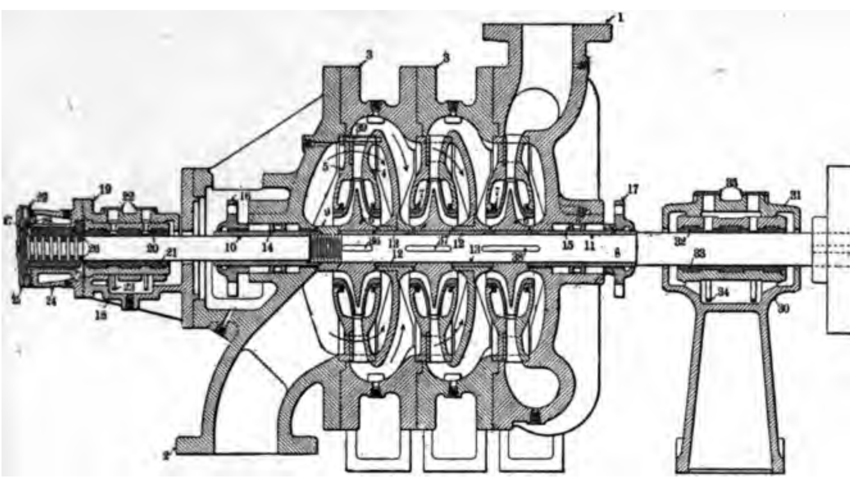


FIG. 125.—Section of double-suction three-stage centrifugal feed pump.

horsepower, and this consumption would not increase with age. The only defect in the centrifugal feed pump is that 200 gal. per minute is about the smallest size which will pay for manufacture, the three common sizes being 500, 750 and 1,000 gal. per minute. The price of the centrifugal feed pump with the turbine drive is about 50 per cent. in excess of the reciprocating pump of the same capacity, and the maintenance of the pump valves in the reciprocating pump will usually be larger than the additional fixed charges. All centrifugal feed pumps should be provided with a check valve on the discharge side, with a bypass provided with a spring valve between the discharge and suction, for use when all of the feed valves happen to be closed, and with a pump governor of some type, which will slow down the turbine when much water is not required.

Cost of Feed Pumps.—The cost of feed pumps is a small item in the cost of the station, varying from 20 to 50 cts. per kilowatt of station capacity, or from 15 cts. to $1 per boiler horsepower the lower prices

Cost of Feed Pumps (Installed) per Engine Horsepower

Simple non-condensing:									
Engine horsepower	10	12	14	15	20	30	40	50	75
Cost per horsepower	$5.70	$5.50	$5.50	$5.40	$4.50	$3.80	$3.15	$2.75	$2.10
Simple condensing:									
Engine horsepower	10	12	14	15	20	30			
Cost per horsepower	$5.70	$5.70	$5.70	$5.70	$5.40	$3.80			
Engine horsepower	40	50	75	100					
Cost per horsepower	$3.10	$2.75	$2.10	$1.70					
Compound condensing:									
Engine horsepower	100	200	300	400	500	600	700	800	
Cost per horsepower	$0.95	$0.60	$0.45	$0.40	$0.30	$0.30	$0.30	$0.25	
Engine horsepower	900	1,000	1,500	2,000					
Cost per horsepower	$0.25	$0.25	$0.25	$0.20					

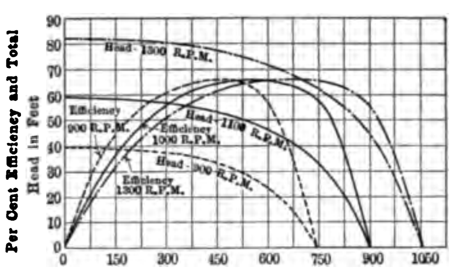


Fig. 126.—Characteristic curve varying conditions, centrifugal pump.

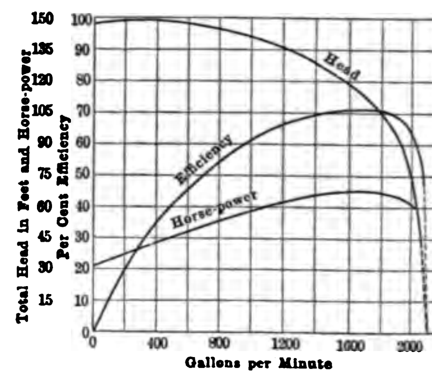


Fig. 127.—Characteristic curve, 8-in. high-speed centrifugal pump.

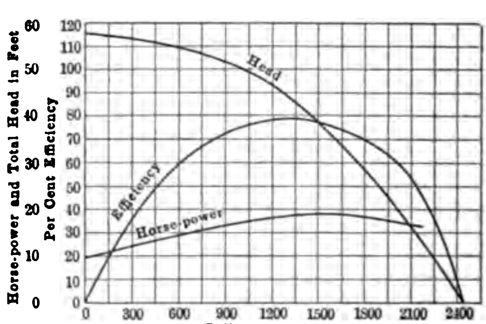


Fig. 128.—Characteristic curve, variable head and constant-speed centrifugal pump.

applying to the larger station. Duplex direct-acting pumps, of the Worthington type, and suitable for boiler feeding, vary in capacity from the 6 by 4 by 6-in., 100-gal. pump to the larger pot-valved pumps with compound steam ends 14 and 20 by 10 by 15 in. delivering 500 gal. per minute. The price varies from about $300 for the small size to around $2,000 for the large size, with the intermediate sizes at proportional prices. Centrifugal feed pumps, turbine-driven, run from 200 to 1,000 gal. per minute, in about four sizes, and cost about $1,500 in the 200-gal. size and about $3,200 in the 1,000-gal. size. Motor-driven centrifugal pumps are about 10 per cent. higher in price, and triplex motor-driven pumps often run from 100 to 200 per cent. higher.

The Injector.—Injectors are sometimes used instead of feed pumps or to supplement them. They have to be carefully adjusted for the steam pressure used. The temperature of feed water supplied to the injector must be below 150°F.

Advantages.—

1. Cheap in small sizes.
2. Compact.
3. No moving parts. No cost for repairs.
4. Delivers water hot to boiler.
5. No exhaust to care for.
6. Delivers warmed water without use of feed-water heater.

These advantages make its use almost universal for locomotives.

Disadvantages.—

1. Stops on reduction of steam pressure.
2. After stopping by failure in steam pressure, often hard to start.
3. Feed water cannot be much over 100°F. in actual operation.
4. Is of little use in large sizes.

The injector uses about as much steam as the feed pump. The injector would seldom be used with boilers above 100 hp. except in locomotive practice where its use saves the installation of feed-water heaters.

Feed-water Heaters.—The exhaust from the feed pump and other auxiliaries can be largely utilized by the employment of feed-water heaters, wherein the feed water is heated nearly to the exhaust temperature by the exhaust steam. The following claims are made for feed-water heaters: for every 11° that the feed water is warmed there is a saving of 1 per cent. in the fuel burned; with sufficient exhaust steam available, cold feed water may be raised to 205°–210°, saving from 10 to 15 per cent. of the fuel. In some localities a heater will pay for itself in a few months, depending on the price of fuel.

Heaters are of two kinds, closed and open. In the closed heater the feed water is pumped through copper tubes around which the exhaust steam is led. The heat of the exhaust steam is transferred through the copper tubes to the feed water and the condensed steam may go either to the feed-water tank or to the sump if it is oily. Closed heaters are rarely used at the present time, unless the exhaust steam is so dirty that the condensate must be thrown away.

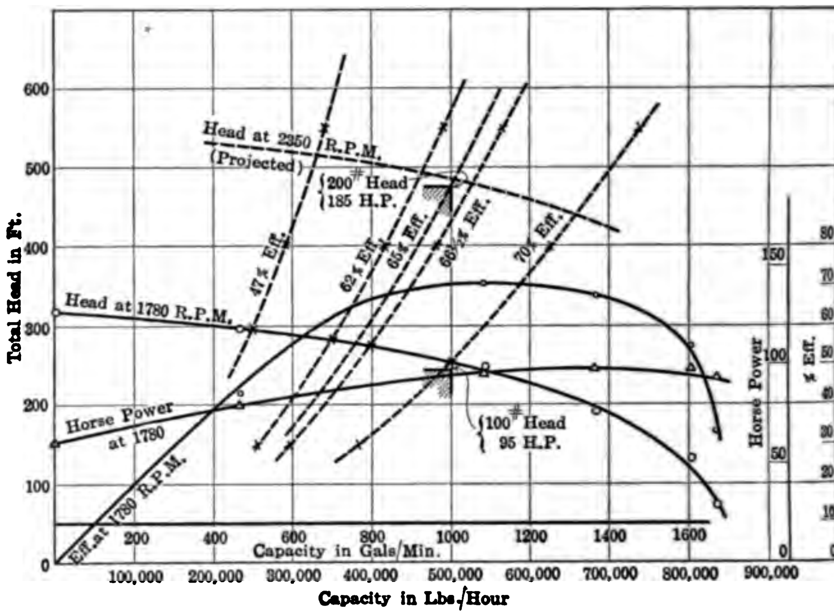


FIG. 129.—Characteristic curves of Westinghouse house and fire service pump.

Open heaters consist of a chamber in which the exhaust steam and feed water are mixed. They usually are of such a size that considerable storage is obtained. Where the feed water is very hard they may be

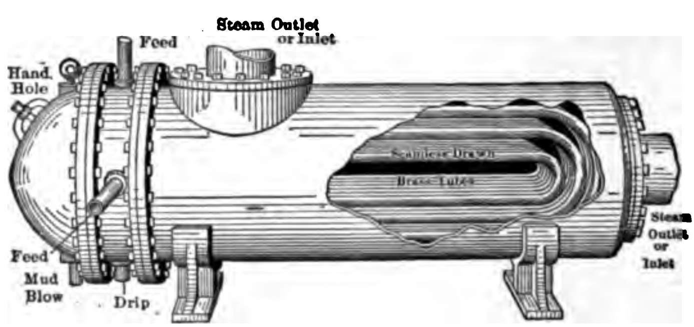


FIG. 130.—Closed heater, Berryman type.

provided with trays on which the carbonates thrown down by the heat are deposited, and frequently they are provided with a sand or excelsior filter which catches other impurities. These heaters, when furnished with this apparatus, might be termed heater purifiers and are largely used.

losed heater is practically a surface condenser working at atmos- pressure, while the open heater is a jet condenser, in some cases purifying attachments.

nere is a third type of heater which was introduced a number of ago but has made little headway toward common use. A tray ays are placed in the steam space of the boiler and the feed is delivered into the upper tray, overflowing and being heated by

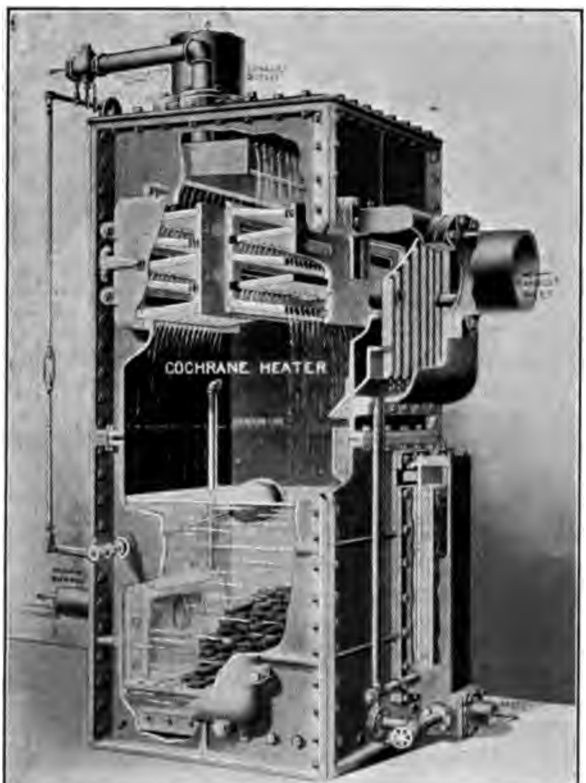


Fig. 131.—"Cochrane" open heater.

team. The impurities collect in the trays, and are removed by what sponds to a surface blowoff pipe. Theoretically there can be no g by the use of a live steam feed water heater. Actually the rted savings are from 1 to 3 per cent. The most valuable feature of type is the surface blowoff and the assurance that no cold water can h the hot boiler surfaces. See Fig. 132.

ost of Feed-water Heaters.—The ordinary closed heater has from o ½ sq. ft. of heating surface for each boiler horsepower, and costs 1 75 cts. to $1 per horsepower.

Cost of Feed-water Heaters Installed per Engine Horsepower

Simple non-condensing:									
Engine horsepower	10	12	14	15	20	30	40	50	75
Cost per horsepower	$2.95	$2.75	$2.70	$2.60	$2.50	$2.20	$2.15	$2.05	$1.90
Simple condensing:									
Engine horsepower	10	12	14	15	20	30			
Cost per horsepower	$2.95	$2.75	$2.70	$2.65	$2.50	$2.30			
Engine horsepower	40	50	75	100					
Cost per horsepower	$2.15	$2.10	$2.00	$1.80					
Compound condensing:									
Engine horsepower	100	200	300	400	500	600	700	800	
Cost per horsepower	$2.85	$2.55	$2.25	$2.00	$1.75	$1.40	$1.10	$1.10	
Engine horsepower	900	1,000	1,500	2,000					
Cost per horsepower	$1.00	$1.00	$0.95	$0.95					

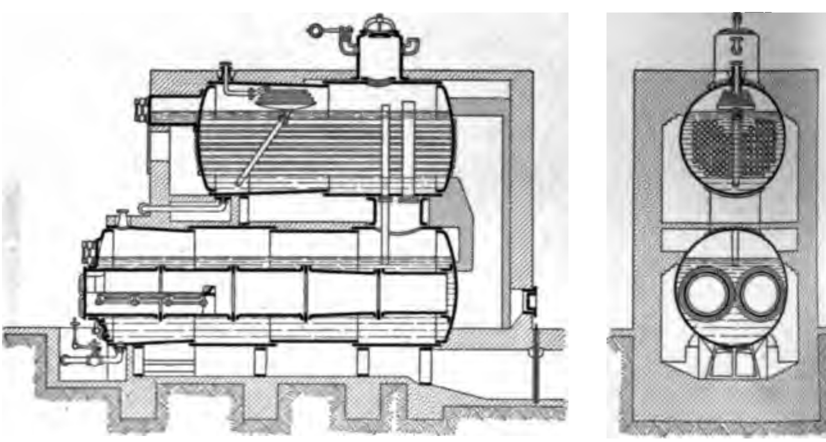


Fig. 132.—Double deck boiler with internal-feed water heater.

Economizers.—Under advantageous conditions, the large waste of heat to the chimney may be very largely reduced by the use of a special form of feed-water heater of the water-tube type, which may be provided with soot cleaners and is located in the flue between the boiler and chimney. Such a heater is called an "economizer."

Economizers are of value in plants operating with steady load in which little exhaust steam is available. The annual maintenance usually amounts to 10 per cent. or more of the original cost. Save under exceptional conditions, boiler heating surface is cheaper and usually gives better results.

One writer states that economizers are guaranteed by the manufacturers to save 6.5 per cent. when the temperature of the water entering is as high as 200°F., the economizers having 4.5 sq. ft. of heating e per boiler horsepower and the boilers working at normal rating. ıl tests show a saving of 10 per cent. with low stack temperatures, a average of 12 per cent. with ordinary stack temperatures.

ıe amount saved would ordinarily pay for the cost of the economizers ›ut 3 years.

ıey cost about $5.40 per boiler horsepower for plants of 1,000 boiler ıd over on the basis of 4.8 sq. ft. per boiler horsepower; or $6 per

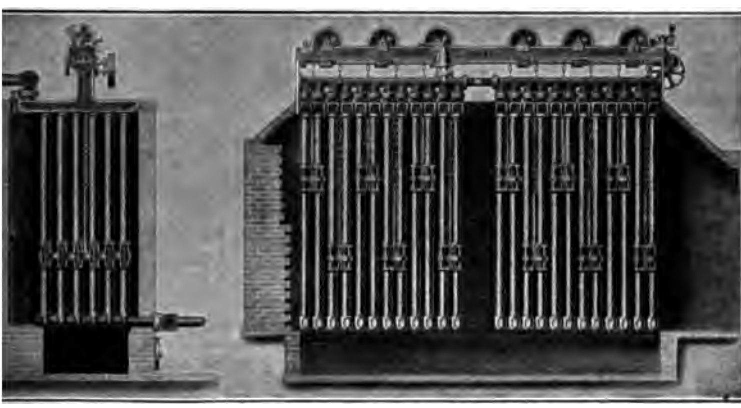


FIG. 133.—Economizer.

horsepower on the basis of 5 sq. ft. This includes the cost of deıg, erecting, brick setting, etc. It is claimed that 3 per cent. of ıvestment will more than pay for the cost of operation, cleaning ›pairs.

ıis same writer gives the following illustrative example:

)0-hp. plant.
ırating 300 10-hr. days per year.
l consumption, 3½ lb. per boiler horsepower-hour.
l, $3 per ton.
ıual fuel cost would be $15,750.
nomizer saving, 12 per cent. = $1,890.
t of economizer, 5.40 per boiler horsepower = $5,400.
r cent. for interest, repairs, operating and cleaning = $432.
saving = $1,458 which is sufficient to pay for the economizer in less than 4

ınt were operated continuously, fuel cost = $45,990.
ıving = $5,085, sufficient to pay for economizer in about one year.

Barrus reports the following results of economizer tests:

Heating surface, boiler, square feet	1,894	1,058	5,592	3,126
Heating surface, economizer, square feet	1,600	1,920	1,280	1,600
Temperature of gases leaving boiler, degrees	376	361	403	435
Temperature of gases leaving economizer, degrees	231	254	299	279
Temperature of feed water entering economizer, degrees	95	79	111	84
Temperature of feed water entering boiler, degrees	175	145	169	196
Increased evaporation produced by economizer, per cent.	10.5	7	9.3	12.8

and W. R. Roney (*Transactions* A. S. M. E., vol. 15) reports:

Plants tested	1	2	3	4	5	6	7	8	9
Gases entering economizer, degrees	610	505	550	522	505	465	490	495	595
Gases leaving economizer, degrees	340	212	205	320	320	250	290	190	299
Water entering economizer, degrees	110	84	185	155	190	180	165	155	130
Water leaving economizer, degrees	287	276	305	300	300	295	280	320	311
Gain in temperature of water, degrees	117	192	120	145	110	115	115	165	181
Fuel saving, per cent	16.7	17.1	11.7	13.8	10.7	11.2	11.0	15.5	16.8

The N. E. L. A. in the June, 1915 Report of its Committee on Prime Movers points out that:

"Exactly how far to push the question of feed-water heating by economizer depends on investment cost, depreciation, operating cost, space required and the economy obtained. The increase in economy due to the economizer described above amounts to from 10 to 12 per cent. If the feed water were supplied to the boiler at the temperature of the steam it would mean an economy of from 16 to 17 per cent., or a further gain of about 6 per cent. This would reduce the temperature of the discharge gases to about 200°F. The cost of accomplishing this must be more than offset by the gain in economy to warrant such an installation —and each case must be considered separately.

If the temperature of the gas is lowered below the dew point, a deposit of moisture occurs on the tubes, with resultant scale and incrustation, and has a further objection with western coals, in that the moisture combines with the sulphur in the flue gas, forming sulphuric acid.

The cleaning of economizer tubes of soot does not as yet appear to be satisfactorily solved. The scrapers as ordinarily fitted have the effect of rolling the deposit on the tubes, making it difficult to remove. Blowing, both with steam and air, has been tried, but generally it has not been successful, or has introduced other difficulties which more than offset its usefulness.

With the present tendency to increase the operating pressure of boilers, it is very evident that some change will be necessary in the materials used in economizer construction, to assure their adoption, even though the economies obtainable are relatively large. With boilers operating at 225 lb., the feed-line pressures may exceed 250 lb., and may even at times reach 300 lb., which is generally considered too high a pressure to be safely sustained by cast-iron construction.

The question of steel construction is possibly not entirely settled though the cost would be materially increased and there is the liability of excessive corrosion due to the chemical properties of certain coals.

PERCENTAGE OF STEAM GENERATED USED BY AUXILIARIES
(Surface-condensing Plants—Steam-driven Pump)

Kind of plant	Feed pump		Circ. pump		Air pump		"Wh"[1]	Steam used, per cent.
	Econ., lb.	Eff., per cent.	Econ., lb.	Eff., per cent.	Econ., lb.	Eff., per cent.		
75- to 500-kw. horizontal direct-acting auxiliaries.	200	80	150	80	150	60	500	12.50
							1,000	15.75
							1,500	19.00
							2,000	22.00
							2,500	24.75
500- to 1,000-kw. horizontal direct-acting auxiliaries.	150	80	100	80	100	60	500	9.00
							1,000	11.50
							1,500	13.75
							2,000	16.00
							2,500	18.25
100- to 600-kw. engine-driven centrifugal circulating pumps, crank and flywheel air pumps, direct-acting feed pumps.	200	80	60	50	50	60	500	8.75
							1,000	11.00
							1,500	13.00
							2,000	15.25
							2,500	17.25
600- to 1,000-kw. engine-driven centrifugal, circulating pumps, drank and flywheel air pumps, direct-acting feed pumps.	150	80	50	52	45	60	500	7.50
							1,000	9.50
							1,500	11.50
							2,000	13.50
							2,500	15.50
Above 1,000-kw. engine-driven centrifugal circulating pumps, crank and flywheel, air pumps, direct-acting feed pumps.	100	80	40	55	40	60	500	7.50
							1,000	7.75
							1,500	8.25
							2,000	10.00
							2,500	11.50

[1] "W" is the number of pounds of circulating water per pound of exhaust steam.
"h" is total head on circulating pump in feet.

Percentage of Steam Generated Used by Auxiliaries
(Jet-condensing Plants—Steam-driven Pump)

Kind of plant	Feed pump		Circ. pump		Air pump		"Wh"	Steam used, per cent.
	Econ., lb.	Eff., per cent.	Econ., lb.	Eff., per cent.	Econ., lb.	Eff., per cent.		
50- to 300-kw. horizontal direct-acting auxiliaries	200	80	150	80	150	70	200	13.00
							300	16.75
							500	20.75
							1,000	25.75
							1,500	30.00
300- to 800-kw. horizontal direct-acting auxiliaries.	150	80	100	80	100	70	200	9.00
							300	11.50
							500	14.75
							1,000	18.50
							1,500	25.00
150- to 500-kw. engine-driven centrifugal injection pumps, crank and flywheel air pumps, direct-acting feed pumps.	200	80	60	50	50	70	200	8.00
							300	9.50
							500	11.50
							1,000	14.75
							1,500	19.75
500- to 1,000-kw. engine-driven centrifugal injection pumps, crank and flywheel air pumps, direct-acting feed pumps.	150	80	50	52	45	70	200	6.00
							300	7.50
							500	9.25
							1,000	12.00
							1,500	16.75
Above 1,000-kw. engine-driven centrifugal injection pumps, crank and flywheel air pumps, direct-acting feed pumps.	100	80	40	55	40	70	200	4.50
							300	5.75
							500	7.25
							1,000	9.75
							1,500	13.75

Radiation and Leakage.—In well-designed plants, with properly covered pipe lines, the radiation and leakage losses may be taken as low as 3 per cent. of the total evaporation, but usually will run much in excess of this figure.

Oil Pumps.—Small duplex steam pumps are usually used for feeding oil to the burners. They are exceedingly uneconomical and a steam consumption of over 200 lb. of steam per indicated horsepower-hour may be considered a fair average. The efficiency of these pumps is also very low, varying between 40 and 50 per cent. It is customary to assume the steam consumption of the oil pumps as being 1 per cent. of the total evaporation, which is conservative. Centrifugal oil pumps have been used to some extent and are much less inefficient and somewhat higher in cost.

Oil Burners.—There is a great variety of oil burners on the market, some of which have given good satisfaction. They may be divided into hree classes, depending on the atomizing agent used and the method of ts mixture with the oil.

1. Burners using steam for atomizing.
2. Burners using high-pressure air for atomizing.
3. Burners using low-pressure air for atomizing.

Practically all commercial oil-burning installations on land use a burner of the first type, as it is simpler, more convenient and more economical. These burners may be again divided into those in which the oil and atomizing agent are mixed inside the burner, and those in which they are not mixed until they leave the burner. Oil-firing does not usually meet with the best results, unless large properly shaped combustion spaces are provided, and this is best secured with some of the more modern types of water-tube boiler. The steam used for atomizing the oil at the burners is a direct loss, escaping up the stack as superheated steam. The best results show a steam consumption of about ⅓ lb. of steam per pound of oil, which is something over 2 per cent. of the gross evaporation. In ordinary cases the use of atomizing steam amounts to from 3 to 5 per cent.

Boiler Feed Water.—Although this is a large subject, the essential features, as stated by Shealy in his "Steam Boilers,"[1] are here presented in modified form.

"The waters of our lakes, rivers, springs, and underground streams contain more or less mineral substances that have been dissolved by the water in its passage through the earth, and also more or less dirt, mud, and vegetable matter which have been taken up and carried along by the water. When water is evaporated in a boiler, all of these impurities are left behind and are usually deposited in solid form. In some cases these substances merely settle as a soft mud and can be blown off, but more often they form a hard scale on the heating surface, which is difficult to remove. The scale thus formed is a very poor conductor of heat and its presence, therefore, reduces the efficiency and capacity of a boiler by reducing the amount of heat that can pass through the heating surface.

It is much better, as far as possible, to prevent the scale-forming substances from entering the boiler, as, once inside, they will form a more or less hard scale which must be removed. Even though the scale formed is soft and easily removed, its presence involves a certain expense in laying off the boiler and cleaning it. To prevent the formation of scale, requires a knowledge of the chemistry of feed water and of the proper treatment by which the mineral salts may be removed before feeding the water into the boiler, or they may be changed in nature so that they will not form a hard scale but will settle as a soft scale or as mud which can be blown off or easily removed.

Impurities in Feed Waters.—The impurities most often found, and found in the largest quantities, are given below together with their chemical formulæ:

[1] "Steam Boilers," E. M. Shealy, McGraw-Hill Book Co.

Calcium carbonate	$CaCO_3$
Magnesium carbonate	$MgCO_3$
Calcium sulphate	$CaSO_4$
Magnesium sulphate	$MgSO_4$

The impurities less frequently found and in smaller quantities are:

Iron carbonate	Fe_2CO_3
Magnesium chloride	$MgCl_2$
Calcium chloride	$CaCL_2$
Potassium chloride	KCL
Sodium chloride	NaCL

Besides these there may be iron oxides, calcium phosphate, silica, and organic matter, which usually occur in very small quantities.

The Carbonates.—Calcium carbonate and magnesium carbonate do not dissolve very readily in pure water, but most water contains carbonic acid (CO_2) and if this is present, the carbonates dissolve very readily. The carbonates unite with the carbonic acid and form the bicarbonates of calcium and magnesium, which are very soluble. This combination can, however, be broken up by heating, which drives off the carbonic acid gas and returns the carbonates to the insoluble form, when they will be deposited. The action described above, begins when the water is heated to 180°F. and by the time it has reached 200°F., the greater part of the carbonates will be deposited. It requires a temperature of about 290°F. to deposit all of the carbonates, but the larger part is deposited between the temperatures of 180° and 200°F.

If the feed water enters the boiler at a temperature lower than 180°F. the carbonates will be deposited inside the boiler, but, if some device is used whereby the feed water is heated to a temperature of about 210° or 212° before it enters the boiler, there will be very little of the carbonates deposited in it and it will be easily cleaned.

The Sulphates.—Calcium sulphate and magnesium sulphate are the most troublesome impurities as they form an exceedingly hard scale which is difficult to remove. The solubility of calcium sulphate in grains per gallon is given in the following table.

Temperature, °F.	Solubility, grains per gallon
212	125.0
300	40.0
350	15.5
400	12.0
450	11.0
500	10.5

When other salts are present the solubility is somewhat larger. Live-steam feed-water heaters will usually throw down a portion but chemical means must be taken to prevent scaling in the boiler as the water becomes concentrated.

The sulphates possess very active cementing qualities, and not only form a very hard scale themselves, but become mixed with mud and other sediment, cementing it also into a very dense, hard scale. The best and cheapest chemical for this purpose is carbonate of soda, which is also known by the names of soda ash, soda crystals, sal soda, washing soda, Scotch soda, concentrated crystal soda, crystal carbonate of soda, black ash, and alkali. At temperatures above 200°F., carbonate of soda or soda ash acts on the sulphate of calcium and magnesium, and also sodium sulphate. The carbonates thus formed become insoluble and deposit at this temperature. The sodium sulphate thus formed remains in solution and passes into the boiler where it gradually accumulates in the water till it can hold no more, when it is deposited. Before it begins to deposit, however, the boiler may be blown down and refilled with fresh water. The Hartford Steam Boiler Inspection and Insurance Co. states that with an average water, such as that of Lake Michigan, requiring 1 lb. of soda ash per 10-hr. day for a 75-hp. horizontal return tubular boiler, the boilers should be blown down two gages every 12 hr., and should be emptied and refilled with water not less than once in 3 weeks.

Chlorides.—Magnesium chloride gives trouble because of its cementing properties. The other chlorides such as calcium, sodium and potassium give little trouble from incrustation unless allowed to concentrate until the water will hold no more in solution, when they are deposited and increase the bulk of the scale. Magnesium chloride is generally supposed to have a corrosive action on the steel plates of the boiler as it reacts with the water, under the influence of heat, forming magnesium hydrate and hydrochloric acid, the acid then attacking the metal of the shell and tubes.

Preventing Scale.—The formation of scale and the troubles caused by it have already been explained. The feed water should be analyzed and steps taken either to prevent the scale-forming elements from entering the boiler or to cause their deposit within the boiler in a form that will not adhere to the metal but can readily be blown out. If scale has already formed within a boiler, chemicals should be introduced to soften it and it should then be removed by washing, if softened sufficiently, or if not, by mechanical means. If the scale is very hard and flinty, it indicates that there is a considerable percentage of the sulphates present. The carbonates form a very soft scale.

Foaming and Priming.—A boiler is said to foam if the steam space is partially filled with unbroken bubbles of steam, and to prime if the steam carries water with it from the boiler.

Foaming is caused by any materials, either dissolved in the water or suspended in it, which retard or interfere with the free escape of steam from the water in the boiler. A collection of scum on the surface of the water is also a common cause of foaming. Scum may be caused by oil, vegetable matter, or sewage which collects on the surface of the water, forming a coating which is hard for the steam bubbles to break when they rise to the surface. If the water contains an alkali, and any animal or vegetable oil becomes mixed with it, the alkali will change the oil into soap, which forms suds and causes foaming. In many power plants the exhaust from engines or pumps is condensed, collected into hotwells, and fed back into the boilers. If the cylinders are lubricated with animal or vegetable oil, there is danger of its getting into the boiler and causing foaming. For this reason, only a mineral oil should be used in the cylinder but, even with this, great care should be taken to prevent its entering the boiler, as it is a frequent cause of burned plates.

Oil extractors placed in the exhaust pipe will aid in removing oil. Open feed-water heaters are usually provided with oil extractors, and feed water taken from such heaters is almost entirely free from oil.

Foaming may also be caused by the concentration of certain salts in the water.

Priming is, in general, caused by the following conditions, all of which should be looked after:

1. Failure to blow down regularly and sufficiently (chief cause).
2. Failure to clean the boilers regularly.
3. Presence of oil, alkalies or vegetable matter.
4. Type of boiler.

Corrosion.—Corrosion is most often caused by the presence of a free acid in the feed water. The free acid may result from the supply of water being contaminated, from adulterants in the cylinder oil which find their way into the boiler, or from the splitting up of certain salts in the water.

All water contains more or less air, which is liberated when the water is heated and which attacks metal surfaces. Air absorbed in water is more active in attacking metal than free air. This is probably due to the fact that more oxygen than nitrogen is absorbed by the water.

The ordinary ingredients of scale, carbonate and sulphate of lime, have little or no direct corrosive action unless the scale becomes too thick and causes overheating. In fact a slight coating of these salts acts as a protection and, in some cases, when the water fed into the boiler is exceptionally pure, the interior of the boiler may be lime washed at cleaning time with advantage.

Another frequent cause of pitting and corrosion is a galvanic action which goes on in some boilers, particularly in marine practice. This may be stopped by placing pieces of zinc in various parts of the boiler. The zinc will be eaten instead of the steel and, therefore, will need replacing frequently.

Treatment of Feed Waters.—In case the feed water is known to contain impurities, a sample of it should be submitted to a chemist who makes a specialty of analyzing feed water, for analysis and prescription for the remedy of be applied. This course should also be followed in the case of a new plant. When the location for a new plant is to be chosen, particular care should be taken to secure a sufficient supply of good water.

The term "good" as applied to feed water is only relative, but the following designations are generally used, based on the number of grains of scale-forming substance in each gallon of the feed water:

Less than 8 gr. per gallon	Very good.
From 8 to 12 gr. per gallon	Good.
From 12 to 15 gr. per gallon	Fair.
From 15 to 20 gr. per gallon	Poor.
From 20 to 30 gr. per gallon	Bad.
More than 30 gr. per gallon	Very bad.

Water containing as much as 20 to 30 gr. of scale-forming materials to the gallon should never be used unless the water is first purified.

For convenience of reference, the different impurities to be found in feed water and the remedies to be applied are collected in the following table:

Troublesome substance	Trouble	Remedy
ent, mud, clay, etc.	Incrustation	Filtration, blowing off.
ly soluble salts.	Incrustation	Blowing off.
onates of lime, magnesium and iron	Incrustation	Heating feed. Addition of caustic lime.
ate of lime.	Incrustation	Addition of carbonate of soda or barium chloride.
ide and sulphate of magnesium.	Incrustation and corrosion	Addition of carbonate of soda.
nate of soda in large quantities.	Priming	Addition of barium chloride.
.	Corrosion	Alkali.
ved carbonic acid and oxygen.	Corrosion	Heating feed water. Addition of slacked lime.
e (from condensed water).	Foaming	Slacked lime and filtering. Carbonate of soda. Substitute mineral oil.
ic matter (sewage).	Priming	Precipitate with alum, or ferric chloride and filter.
ic matter.	Corrosion	Precipitate with alum, or ferric chloride and filter.

The Permutit water purification process which has been recently oduced depends, for its action, upon the power of "base exchange" essed by zeolites. The process consists of pumping the raw feed er through a tank containing an artificial zeolite made by fusing kao- feldspar, pearlash and soda. This material is broken up into small es and packed in the shell. The calcium and magnesium compounds converted into sodium compounds which are very soluble. When the mutit becomes exhausted, it is regenerated by a strong solution of mon salt which is allowed to remain in contact with it for about 8 hr. s process is not practical with a large amount of lime salts in solution he cost is prohibitive. Cold processes for water softening are quite ely used and with bad waters are beneficial but are somewhat costly cost of treating may vary from ½ ct. per 1,000 gal. to as high as 7 cts. per 1,000 gal. Where the Permutit process may be used its should not exceed 4 cts. per 1,000 gal. Hot processes are best worked e open feed-water heater which should be provided with large water- age capacity and filtering arrangements.

Soot Blowers.—In modern installations of both water- and fire-tube ers, soot blowers are regularly installed as a part of the permanent pment and are used as often as may be necessary to secure good opera- . These tube cleaners consist of a set of nozzles so set that all the s of a fire-tube boiler can be cleaned at once. In water-tube boilers t of nozzles are provided for each pass and the nozzles are so arranged practically the whole heating surface may be covered. Steam ir is used and the tubes are cleaned while the boiler is in light service. installation cost varies from 50 cts. to $2.50 per boiler horsepower.

Fig. 134.—Vulcan soot blowers applied to Babcock & Wilcox boiler.

PROBLEMS

50. The owner of a maufacturing plant is about to install a 400-hp. compound con-densing Corliss engine with surface condenser, water-tube boilers and the necessary auxiliaries. The cooling tower will be placed on the roof of the building, 70 ft. above the pump pit.

1. How much steam ought the boilers to supply per hour when the engine is oper-ating at full rated load?

2. He proposes to install three boilers of equal size, two of which shall supply the steam demand in (1) when operating at 25 per cent. above the manufacturer's rating. How many square feet of heating surface should each boiler contain?

51. Determine the amount of steam used by a feed pump per hour for each plant indicated in problem 18, page 81.

CHAPTER X

PIPING

Although the choice and arrangement of certain generators and prime movers determine in a general way the efficiency under which a generating station can work, yet the piping systems may influence this result to a much greater extent than is generally believed.

The size and arrangement of the various pipes and valves have a very mportant influence on the efficient and economical operation of the lant. The piping system may be classified under the following heads:

High-pressure steam piping, main and auxiliary.

High-pressure drip piping and boiler returns.

The feed-water piping (high pressure).

The exhaust piping.

The circulating water piping for condensation.

The hotwell and low-pressure drip piping.

The make-up feed piping and city water supply.

The jacket and wetting-down piping.

Compressed-air piping.

Oil piping, both low-pressure for lubrication and high-pressure for ep bearings, and

The fire lines, which are ordinarily considered part of the plumbing ontract and put in separately from the pipe job.

In the design of these various systems consideration must be given o drainage and returns (traps and steam loops), expansion bends, slip joints, etc., vibration, angles and supports, and the various materials which are proper to use for the different purposes which the piping systems must fulfill. The joints or flanges, the gaskets to be used between the flanges, the design of the fittings, elbows, tees, etc., the types and designs of the valves, must all be considered in connection with each class of piping.

Pipe.—The material for steam pipe, whether high or low pressure, is now almost uniformly openhearth steel. This may be made by the acid or basic process, but Bessemer pipe or wrought-iron pipe should not be used if the best results are to be obtained. The use of Bessemer-steel pipe brings in difficulty in flanging and bending is usually uncertain at the welds. It has in its composition rather more phosphorus and sulphur than is considered good when severe strains are to be placed on the material. Wrought-iron pipe, when it can be obtained, may be very good for certain uses but it is almost impossible to flange a piece of wrought-iron pipe satisfactorily and its use is now confined mainly to unimportant work at localities close to the place where the pipe is made. Steel pipe when used for oil or salt water is often galvanized and its thickness should be proportioned to meet the pressures in use. Where warm water is to be distributed, cast-iron pipe has been and is the standard. Cast-iron pipe, when properly made, has proved to be the best for large and small water mains for either low or high pressures. Where the water pipe is small or where many bends are required, or where the heat and wear are excessive, bronze pipe has been substituted for cast-iron with very good results. The smaller sizes of pipe used in the oiling system are almost invariably made of brass. The use of copper pipe for steam work has been almost entirely superseded, the introduction of superheated steam with the resulting action of the high temperature on the copper rendering it unfit for such employment. There are many stations in which nothing but openhearth steel and cast-iron piping are used and it may be noted that this practice is increasing and these materials will be the standard for the future.

Joints.—Pipe joints have been a great source of trouble in the past and the various kinds and "standards" have been as many almost as there were individual engineers. For low-pressure pipe work the screwed joint with the standard pipe thread and cast-iron flanges has been and is the standard for the best work. For pressures above 100 lb., however, another type of joint should be adopted if the best work is desired. For this purpose there has been no joint found better than the so-called Van Stone joint. This is made by flanging the end of the pipe against the outside of a steel or cast-iron flange. There are many varieties of welded flanges in which the flange is welded directly to the pipe, but these do not seem to have been as popular or as good in construction as the so-called Van Stone, although many people use them. All welded flanges have the disadvantage that they cannot be turned on the pipe, making great care necessary to avoid mistakes in drilling them.

The best joints are made by grinding the seats to a perfect surface and then bolting them together without a gasket. This, however, takes a high grade of mechanic and has been satisfactory only when made in the proper manner. Instead of grinding the faces, it is now considered at least as good to fine tool finish them and insert a gasket which in the best work has been made of very soft steel approximately $\frac{1}{100}$ in. thick. Duralite and other indurated fibers make good gaskets. Copper gaskets appear to deteriorate very rapidly in this position and are not used on high-pressure work as much as formerly. The tongue and groove joint cannot be recommended for steam work as it is almost impossible to ring two joints to the same degree of tightness. For the lower steam ressures copper gaskets work very nicely and are now standard. These re usually stamped with corrugations which flatten out when the bolts re tightened up, assuring a surface practically the whole width of the ace. For exhaust work rubber with wire insertion such as the "Rainow" is mostly used. For water, whether hot or cold, the "Common ense" or other babbit composition gaskets are quite satisfactory. The asket made up of a soft lead ring with a copper wire ring outside of it as also been largely used with very good results and "Rainbow" gaskets re satisfactory when the pressures are not too high.

Fittings.—For low-pressure work, either steam, exhaust or water, the Master Steam Fitters' Association has adopted a standard of pipe fitting which is practically used throughout the United States and it is only for ressures higher than 300 lb. that special fittings are required. Up to 00 lb. steam pressure with no superheat, cast iron or gun iron forms the leal material for pipe fittings and is practically the only material in use. Vith the advent of superheated steam, however, the cast-iron fittings oon proved themselves to be useless with the high heats and semi-steel and steel fittings were tried with the best of results. Today no plant ising superheated steam installs cast-iron fittings for high-temperature work. All fittings should be provided with proper means of draining and drainage pockets or outlets should be placed at the lowest points for the attachment of the drainage system.

Valves.—For low-pressure work the standard cast-iron valves with bronze seats have been more than satisfactory. They are now made of a great many types, all of which give very good results. The solid wedge gate is perhaps the earliest and the best known. The spilt-wedge type and the parallel two-gate type are also well known and largely sold. Globe valves are not usually used for steam work on account of the resistance offered to the passage of the steam, but for throttle valves and for stop valves are still the standard. For high-pressure work and especially with superheated steam the use of the steel-body valve with steel seats and discs has become standard and many varieties of valves are now on the market, some of which are doing excellent work. Nickel-bronze and nickel are also used for seats and stems with good results. In choosing a valve for high-pressure and high-temperature work, great stress should be laid on the absence of chance for unequal expansion in the body and gates. The metal should be so placed that what expansion occurs will be equable in all directions and the gates so designed that they cannot spring out of true under different degrees of heat. Such mechanism as may be used between the gates in a double-gate valve to press them up against the seats should be as carefully designed as the body of the valve, as small deflections in this part of the mechanism will prevent tightness.

The most satisfactory valves for this work have been of the double-wedge type, although there are good parallel seat and solid-wedge valves on the market which have stood severe tests.

Bolts.—It has been customary in ordinary work to use the standard sizes of bolts as provided by the Master Steam Fitters' Association, using a number of bolts of reasonably small diameter. These bolts are the ordinary iron or Bessemer-steel stock with square heads and semi-finished hexagonal nuts. If these bolts are made of good openhearth steel and are set up in the proper manner, it is an insurance against troubles in the pipe joints whereas the ordinary bolt will probably stretch enough to cause more or less trouble, not to say anything of the action of the hot steam upon the bolts. A leaky pipe is more than a nuisance, for when it has persisted for some time it means that the flanges must be refinished before a tight joint can be obtained.

High-pressure Steam Piping, Main and Auxiliary.—The high-pressure steam piping of a power station consists first of a steam line taking the steam from the boiler drums and delivering it into, second, the first steam main or steam header, and third, the connections from the steam header to the various prime movers. The auxiliary steam may be taken from the boiler drums into the auxiliary header with connections to the auxiliaries, or the connections to the auxiliaries may be made from the main itself, or from the connections to the prime movers. All of these systems are in satisfactory use.

The type of the main steam lines, however, depends very largely upon the layout of the station. With the end-to-end layout, in which the boilers are placed at one end of the station and the prime movers at the other end with main steam line connecting the two, the piping may be likened to a tree, the boiler piping being the roots, the trunk of the tree the steam main and the branches of the tree the feeders of the prime movers.

This type of station is very rarely built at the present day on account of the sizes necessary for the steam main through which all of the steam must pass. For stations larger than about 5,000 kw. it is never used and the ordinary arrangement is the back-to-back where the boilers are arranged in one line and the prime movers in another parallel line, with the steam line parallel to both and between them. The boiler connections then are taken directly to the main and the leads of the prime movers directly from the mains to the prime movers. This system is modified into a unit system by leaving out the steam main and by introducing small equalizing pipes between the unit lines connecting a group of boilers with the prime movers. Further modifications of this layout were brought out by the use of the double-decked station in which the prime movers are placed in the basement with the boilers overhead or the boilers

: placed in the basement with prime movers overhead. Stations of th of these types have been in operation for a long time. There are o stations with the prime movers in the center and a double line of ilers on either side of the engine room. This brings in complications the steam piping, but is frequently economical in cost.

Disadvantages of Various Systems.—The first system described, or e trunk system, is quite a costly system to install and as all of the eam had to pass through one section of the main it necessitates very rge pipes and the consequent serious increase in the cost; the expansion fficulties are magnified by the length of the main steam line and except r very small plants it is no longer used.

The parallel system is probably the most used of all systems and the iit system and ring-header system are modifications of it. In the ring-ader system the steam main is a continuous ring which may extend ound the prime movers, or may be simply a loop between the prime overs. The unit system is frequently turned into a ring-header system th smaller pipe connections between the parts of the unit system. gs. 135, 151, and 157, show various steam-pipe layouts of these types.

The unit system almost invariably presents the cheapest and most tisfactory system of piping, the steam lines being smaller in size. suffers the disadvantage that when the prime mover is out of service e boilers connected to it are also out of service. The parallel system is obably more popular than the unit system and is very frequently used. he cost of this stands next to the unit system and is about the same as e unit system with the cross-ties, which, in reality, convert the unit stem into a parallel system. The ring-header system is probably the ost used and is without doubt the most flexible, but is also much more stly to install on account of the double line of main headers which are sually the largest steam pipes in the station.

There is a great difference of opinion among engineers as to the best ethod to be followed in these layouts. Formerly it was considered ecessary to install a duplicate system of steam mains, each boiler having connection to each of the two mains and each prime mover con-ections to both mains also. This led to a complete duplicate set of eam piping which was extremely costly and usually gave a great deal f trouble. It is very easy to keep a steam line tight when it is hot all e time, but a steam line first hot and then cold is usually much more oublesome. In cases where the duplicate system was installed, one f the lines has been removed and the stations are now running on a ngle system with much better results. The losses in a steam piping stem are entirely due to radiation, the drop in steam pressure due to the iction in the pipe being manifested in heat and radiated from the surface f the pipe. It is now considered the best practice to make these pipes

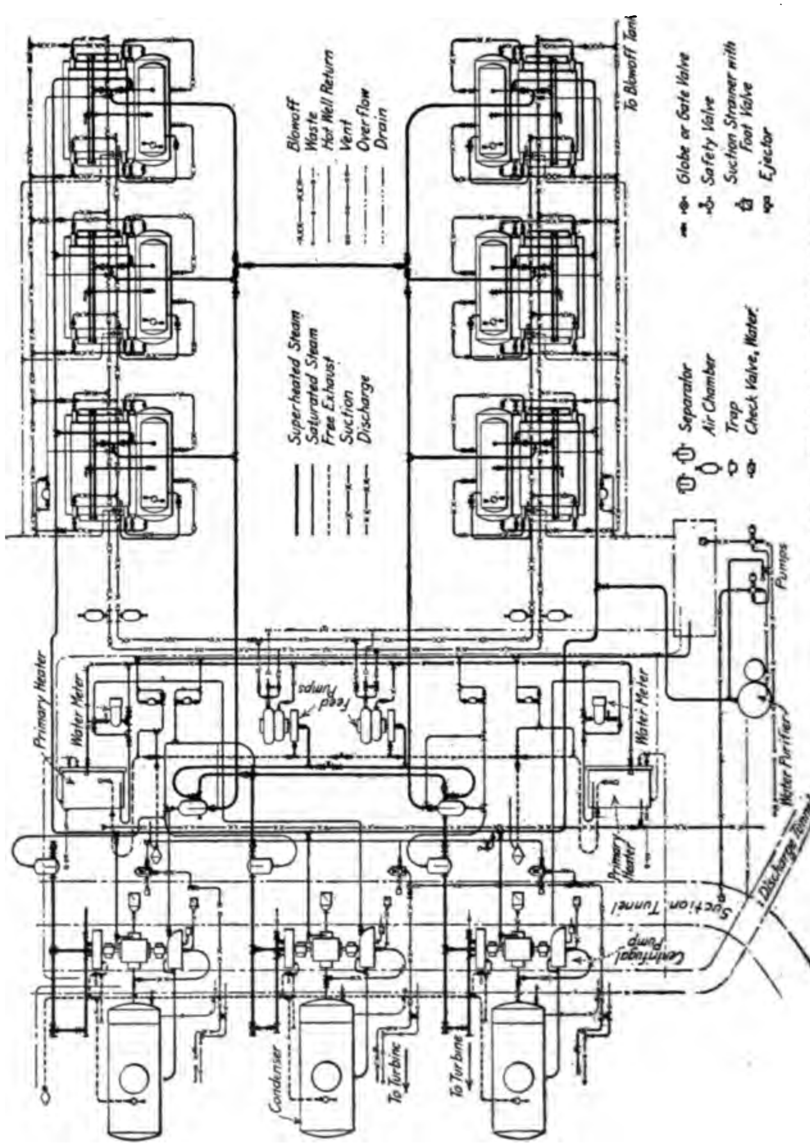


FIG. 135.—Piping diagram of the Eberswalde station, Markische Electricitatswerke.

s small as possible allowing at least from 3 to 5 per cent. drop between he boiler drum and the prime mover. By cutting down the size of the pipe the surface is reduced and as the radiation is directly proportionate to the surface large savings are made.

Formerly steam mains of 18 to 24 in. in diameter were considered necessary in small stations and the radiation owing to the poor quality of pipe covering then used was enormous. At the present time very few mains are put in of larger size than 14 in. I.D. and in some cases a 10- or 12-in. main is considered sufficient. The drop in pressure between boiler and prime mover is considerably larger than formerly but the actual heat loss due to friction and radiation is very much less than it was. This reduction in pipe sizes also brought in great economies in the upkeep cost of the steam lines as with the high pressures carried at the present day it would be almost impossible to keep a 20-in. or 24-in. main tight under the conditions of actual service.

Steam Speeds.—For years standard practice for the speed of steam in steam lines was 4,000 ft. per minute as the minimum, 6,000 as the average and 8,000 as the maximum. This was considered the standard in the days when 125 lb. steam pressure was carried without superheat. At the present time with 200 lb. pressure and superheat which may extend as high as 200°F. above the saturation temperature, the minimum steam speeds are much higher and very few engineers are using as the minimum speed less than 8,000 ft. per minute, the maximum in some cases running as high as 18,000 ft. with no bad results.

Details.—Starting from the boiler it is good practice today to connect the various boiler drums with what is known as a crossover header consisting of a steel casting with ball-and-socket joint connections to the various boiler drums and provided with a single outlet at the top from which the steam supply may be taken. This header is connected to the boiler drums by means of cast-steel nozzles which are riveted to the drums and which have on their upper flanges a ball-and-socket joint. On top of this crossover is placed some variety of automatic stop check valve which is required by the police regulations of certain cities. This stop check valve is made in many styles and performs a variety of functions. It is usually so arranged that when the pressure in the boiler drops below the main pressure the valve will shut preventing the steam from returning to the boiler. It is usually provided with an automatic closing device so that when a steam pipe breaks and a sudden drop of pressure occurs in the main the valve will also shut. It is also provided with a hand closing and opening device. Such valves are shown in Figs. 92 and 136.

From the outboard flange of this valve the main boiler connection is taken to the main. This is of bent steel pipe with Van Stone flanges and is not usually larger than 10 in. for a 650-hp. boiler. Of late the sizes have been cut down to 8 in. and in some cases to 6 in. with very good results. Between this connection and the steam main a gate valve is always placed so that there may be two valves between the boiler drums and the steam main. This is good practice apart from the police regulations which usually require that every connection to the boiler shall have at least two valves between the boiler and the main.

The steam main itself is divided into sections by means of gate valves some of which may be provided with hydraulic or electrical closing and opening devices so that they may be operated from a distance if necessary in cases of emergency. But all valves of this type should be provided with hand closing and opening gear as well as the mechanical gear. From the steam main at a convenient point the connection to the prime mover is made. At the steam main a gate valve is located and the lead is taken by the most direct methods with large bends to the throttle valve of the prime mover. It is not usual to place a second valve between the gate valve and the throttle on the prime mover as the automatic throttle and the throttle valve itself are considered as giving sufficient safety.

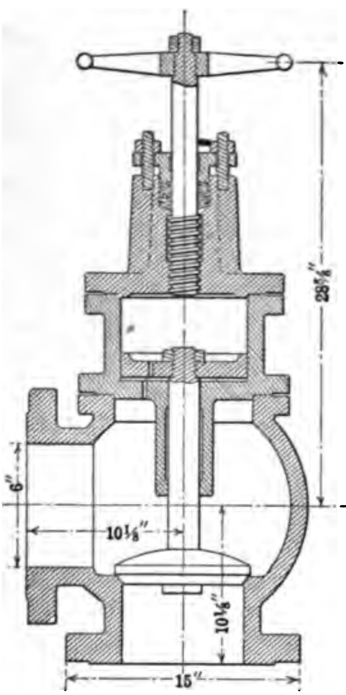


FIG. 136.—Automatic-stop check valve.

Auxiliary Steam Piping.—It was formerly considered the best practice to have the auxiliary steam piping entirely separate from the main steam line and to this end a separate connection was made with a small valve to the end of the crossover pipe on every boiler, these connections being led into an auxiliary main of smaller size extending across the rear of the boilers. This was an entirely separate system connected in no way with the main system. It has the customary two valves between the boiler and the main and a single valve at the main where the connection is made to the auxiliary, the throttle valve of the auxiliary engine acting as the second valve between the engine and the main.

It is now considered better practice that the steam connections with those auxiliaries that are intimately connected with a prime mover should be taken directly from the steam connections of the prime mover near the throttle valve. In many of the latest stations this scheme has been carried out with very good results. This means, however, that a separate system must be provided in the boiler house for the feed pumps, fire pumps and other boiler room auxiliaries which is usually done by means of a similar separate piping system either taken from the boiler drums or else from certain points on the steam main that must always be in service. This is without doubt the best system where superheat is used in both prime movers and the auxiliaries.

High-pressure Drip Piping and Boiler Returns.—It is customary to install a drip system along the under side of a high-pressure steam main to return the water of condensation to the drip tank or boiler. This system is usually built up of small pipe with screw joints, pipe not over 2 in. size being used for this purpose. Every fitting and valve in the main is tapped for a drip connection and a nipple is screwed in with a valve. Similar drips should be installed in the boiler connections next to the main. One of the drips on each boiler connection should be so arranged that it may lead into the ashpit of the boiler where it can be observed and this pipe is left open when the boiler is open for inspection showing that there is no steam next to the stop check valve.

As the high-pressure drips are among the most important pipes in the station it is customary to make these of extra heavy pipe and a great deal of care is usually taken that all of the joints are tight and that the system is a substantial one in every respect.

Feed-water Piping (*High Pressure*).—The high-pressure feed-water piping consists of all of the piping connecting the feed-water pumps with the boilers themselves. This includes the lines through the closed heaters when they are installed and also lines to and from the economizers. This pipe is almost invariably made of cast iron and for ordinary work does not exceed 8 in. in diameter even in the largest stations. Such pipe is usually from 3/4 to 1 in. thick with heavy flanges and raised seats. This piping system usually consists of a run of piping under each row of boilers from which a loop is taken up over each battery and down again connecting with the main on the other side. This loop is most always of 3-in. pipe and is provided at a point above the floor with a check valve and sometimes with a hose connection. The stop valves and check valves are usually of brass and the piping running over the boilers and between check valves is usually brass pipe, iron-pipe size, with brass flanges screwed on and sweated. In the middle of the battery above the boilers a gate valve is placed to separate the two parts of the loop, and brass fittings are inserted above the drums to provide for the 2-in. connections to the front ends of the boiler drums. These connections are bent brass pipe, iron-pipe size, and at the drum are provided with a combination stop and check valve so that any line may be thrown out of service if desired. This is not the standard arrangement, however. The standard consists of a double line of 3-in. pipe extending up at the middle of the battery and connecting with a 2-in. line to each of the three drums. This line usually interferes with stoker installations or with the middle column which runs up between the two boilers of a battery and in large stations the installation is almost always made as first described.

Each boiler maker usually has his own type of stop check valve at the drums and this valve is usually furnished with the boiler. The check valves and stop valves and the gate valve in the middle of the boiler overhead are usually also furnished by the boiler contractor. The piping below the main 3-in. stop valves is usually cast iron and is furnished by the pipe contractor. These mains run below the lines of boilers and are connected to the main feed line which may run the length of the station. This line is sometimes made as a ring header or closed loop. Sometimes it consists of a double line of mains with crossovers protected by gate valves which are usually of cast iron.

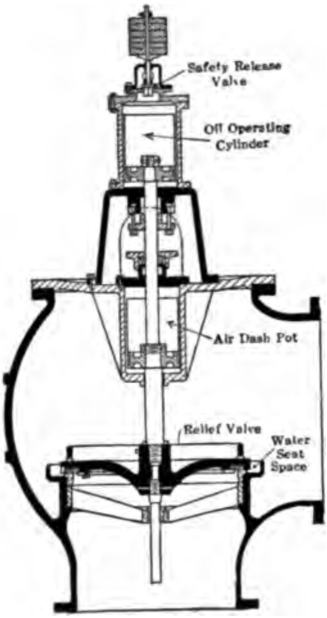


FIG. 137.—Atmospheric relief valve.

Suitable air chambers for equalizing the pulsations are provided when reciprocating pumps are used.

The use of steel pipe for feed-water lines is not to be recommended under any circumstances. The hot pure water affects the material very badly causing pittings which, with certain impurities that are present, will probably destroy the pipe in a very short time. Cast iron seems to stand this sort of work much better than anything else which has been used and is very satisfactory. In many cases where steel has been used it has had to be removed within a very short time and cast-iron pipe substituted.

Exhaust Piping.—The smaller sizes of exhaust piping up to 6 or 8 in. are usually made of standard cast-iron pipe with cast-iron flanges for say 50 lb. pressure. Between 8-in. and 30-in. spiral riveted galvanized pipe with steel flanges riveted on is commonly used; and riveted steel pipe for sizes above 30 in. Allowable speeds of exhaust steam in these pipes are very much larger than are allowable for pressure steam speeds, as high as 35,000 ft. per minute being permissible in certain cases. When a prime mover is arranged to be run continuously the exhaust connection to the condenser is usually made very short and of cast iron. The atmospheric exhaust is connected into this pipe between the prime mover and condenser and is provided with an atmospheric relief valve which is arranged to open wide whenever the vacuum drops to a certain amount. Fig. 137 shows a type of relief valve which has proved satisfactory in service. They are almost always balanced valves of the globe type, provided with weights and dashpots to prevent chattering and arranged for quick and full opening under operating conditions. Hydraulic devices are usually installed to allow opening or closing from a distance. The smaller apparatus, which is always run non-condensing, is provided with a direct exhaust pipe to the heaters where the steam is condensed at atmospheric pressure for heating the feed water. When turbine-driven auxiliaries are used and run non-condensing it is very important that there be no back-pressure at the exhaust nozzle of the turbine and great care should be taken that the exhaust pipes are sufficiently large and straightaway, that no pressure may be developed at the exhaust nozzle.

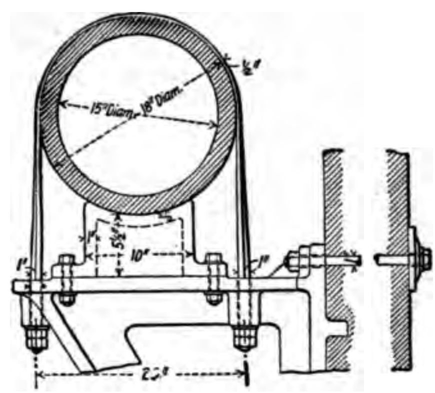


FIG. 138.—Anchor for steam main.

All exhaust piping should be laid out with a fall to the heater or else should be properly graded with drip piping of sufficient size. In a small plant where the exhaust is allowed to go to the atmosphere, suitable mufflers should be installed at the top of the exhaust line to prevent noise. (See Fig. 143.)

Determining Pipe Sizes.—Many formulæ have been used for determining pipe sizes for steam engines, but most of them are now obsolete. In *Power* for Jan. 19, 1915, F. W. Salmon presents results secured by plotting the necessary data from a large number of successful plants.

If A = area of pipe bore in square inches,
d = diameter of pipe bore in inches,
W = average pounds of steam per hour,
C = a constant for a given pressure in the pipe,
K = a constant $= \dfrac{\pi \times C}{4}$,

then $W = A \times C = d^2 \times K$

or $d^2 = \dfrac{W}{K}$.

Salmon presents the following tables for determining suitable pipe sizes:

Constants

		C	K
Vacuum, inches, Hg	28	50	39.2
	26	84	66.0
	24	105	82.5
	22	122	95.7
	20	134	102.0
	18	144	113.0
	16	151	118.6
	13	162	127.0
	6	176	138.0
	0	187	147.0
Gage pressure, square inches	80	267	210.0
	100	275	216.0
	125	284	223.0
	150	291	229.0
	175	298	234.0
	200	304	239.0

Grading of Pipe.—The slope of the pipe line should be toward the engine as water is often prevented from flowing back against the steam current. Drip pockets should be used in all fittings. The main line should pitch about 1 in. in each 10 ft. of run. Pipe lines always need draining. 1- to 2½-in. drain pipes should be used. It is frequently

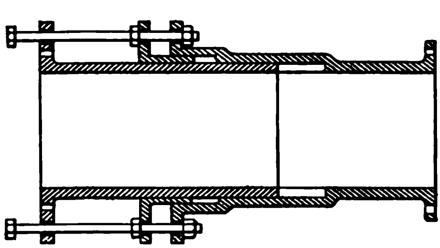


Fig. 139.—Slip expansion joint.

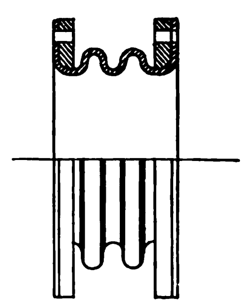


Fig. 140.—Wainwright expansion joint.

convenient to use the drip lines as bypasses instead of bypassed valves in high-pressure work.

Expansion of Pipe.—The coefficient of expansion for ordinary steam pipe seems to be about 0.000006 of its length for each degree F. A rough and ready rule is to allow ½ in. for every 100 ft. for every 100°F. difference of temperature. All pipe lines should be laid out to take care of this expansion, and to this end large radius bends should be employed wherever possible. It is usual to cut the pipe so it will be the right length when it is hot, making up the joints and pulling them together, so that an initial stress is put in the cold pipe. When the pipe becomes hot, this stress disappears and the pipe will then be in equilibrium. Suitable hangers should be provided every 10 or 12 ft. to support the pipe in its proper place. These may consist of a band around the pipe with a rod hanger from some of the floor beams overhead, or the pipe may be supported from below on a roller. Anchors should also be provided at certain places so that the direction of expansion may be controlled. On very long lines sliding expansion joints become necessary, or the corrugated-steel expansion joint may be used.

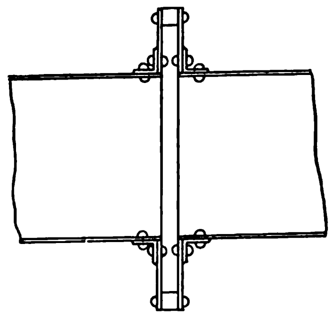


FIG. 141.—Baragwanath expansion joint.

Pipe Coverings.—The best covering is 85 per cent. carbonate of magnesia, 1 in. thick on exhaust piping and 2 in. thick on high-pressure steam piping. This should be covered with ⅛-in. asbestos board and with sewed and pasted canvas. Removable flange coverings should be used. (See paper by McMillan, A. S. M. E., December, 1915.)

Badly erected and leaky lines of steam piping may cause excessive waste. More steam (250 boiler hp.) can leak through a 1-in. hole in a steam pipe at 150 lb. steam pressure than one fireman would usually supply by steady coaling.

Leaks in steam pipe are usually regarded as insignificant but they rapidly dissipate the heat generated in the consumption of a large amount of coal. Uncovered steam pipes also waste large amounts of coal and load the steam with water; water in the steam causes pounding or water-hammer in the pipes, which often produces serious results.

A good steam covering will save some 80 per cent. of the loss of heat which takes place from the naked pipe and the investment in a good pipe covering will usually more than repay 100 per cent. interest.

Cost of Piping.—The cost of piping in a steam power plant varies greatly with the type of installation and with the size of the plant, ranging all the way from $10 to $15 per rated horsepower installed in small plants, to $1.50 to $2.50 per rated horsepower installed in plants of from 3,000 to 5,000 hp., using 125 lb. steam pressure. For turbine plants and engine plants using 175 to 200 lb. steam pressure with superheat, the piping cost for plants of 3,000 to 5,000 hp. will vary from $2.50 to $6. In large turbine plants using high-pressure steam and high superheat the cost may be in excess of $8 per horsepower.

In one engine plant of 28,000 hp. the steam piping system cost $4.35 per horsepower; the feed-water system, 30 cts.; the drip system, 25 cts.; the blowoff system, 10 cts.; the condensing water piping for jet condensers, 30 cts.; the house, fire and heating piping, 15 cts.; the jacket piping, 10 cts.; and the oil system and piping, 75 cts. This system used steam at 175 lb. pressure with no superheat. The use of steel fittings and the careful construction necessary for high-pressure superheated steam work has greatly increased the cost of steam piping. The one item of labor has practically doubled in the last 10 years. The list prices of pipe and fittings should be discounted from 50 to 75 per cent.

Eighty-five per cent. magnesia, 1 in. thick, costs in the neighborhood of 30 cts. per square foot in place. In general such covering costs about one-half list price including labor. One writer states that one man will cover 100 ft. of straight pipe or 40 fittings per day up to 4-in. pipe size. Above 4 in. the cost per 100 ft. of pipe length will be greater owing to the increased labor of handling.

One consulting engineer reports the plant piping cost per indicated horsepower rating of the plant as follows:

Simple non-condensing:									
Engine horsepower	10	12	14	15	20	30	40	50	75
Cost per horsepower	$8.30	$8.00	$7.60	$7.40	$6.70	$5.70	$5.10	$4.60	$3.90
Simple condensing:									
Engine horsepower	10	12	14	15	20	30			
Cost per horsepower	$11.20	$11.00	$10.70	$10.20	$9.50	$8.00			
Simple condensing:									
Engine horsepower	40	50	75	100					
Cost per horsepower	$7.70	$7.30	$6.10	$5.70					
Compound condensing:									
Engine horsepower	100	200	300	400	500	600	700		
Cost per horsepower	$13.80	$11.20	$9.10	$8.00	$7.40	$6.80	$6.50		
Compound condensing:									
Engine horsepower	800	900	1,000	1,500	2,000				
Cost per horsepower	$6.25	$6.00	$5.75	$5.10	$4.55				

Exhaust Heads and Oil Extractors.—The atmospheric exhaust pipe which usually leads into the air above the roof of the station would be very noisy and likely to create a nuisance from the water entrained in the exhaust steam. To avoid the noise and save the water a muffler or exhaust head is fitted to the top of this pipe. This usually consists of a conical or cylindrical chamber two or three times the diameter of the pipe in which are placed wire screens and other baffles breaking up the flow of the steam and the organ pipe effect which would otherwise be produced. Suitable baffles are also installed to catch the entrained water which is piped back to the heater or sump.

Where the condensate from the exhaust steam from engines is to be used over again it is necessary to install a grease or oil extractor in the exhaust line. This usually consists of a set of baffles in an enlarged section of the pipe very similar in design to the steam separator, but of much larger size, as the velocities must be very low in order that the oil may be deposited. It should be remembered that if the oil becomes emulsified or volatilized it will be practically impossible to catch it. In such cases it will be better to change the quality of the oil or throw away the water. There are also grease extractors made to remove the

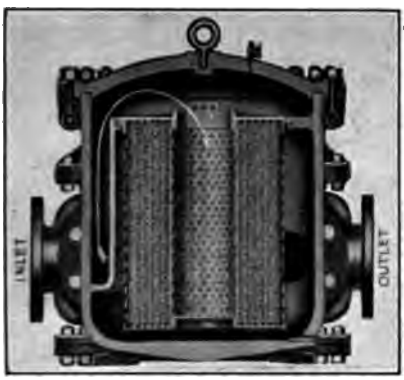


Fig. 142.—"Lagonda" oil filter and grease extractor.

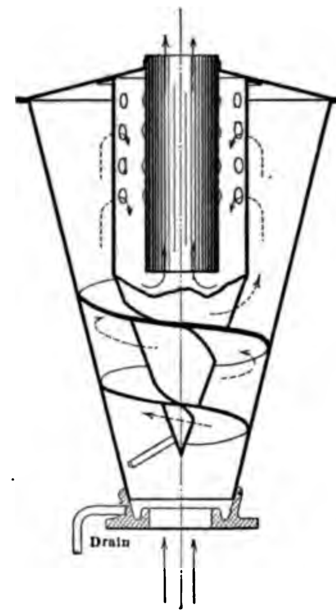


Fig. 143.—Exhaust head.

emulsified oil from water by means of electrical currents. These machines are good in their way, but are costly and take up a great deal of space. Another type of grease extractor is the pressure filter with cartridges of absorptive material, such as excelsior. These machines are also bulky and not much used.

Steam Traps.—The high-pressure engine drips should, if possible, be led back direct to the boiler, but if this is not possible, the next best place is the open heater. The low-pressure drips contain more or less oil, and should be thrown away. If, however, it is necessary to save them, a steam trap may be used which will lift the returns to a sufficient height to enable them to flow by gravity to a heater or storage tank. Steam

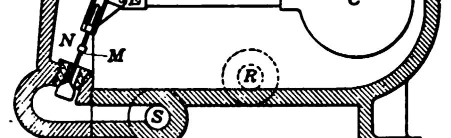


Fig. 144.—Steam trap, float type.

traps may also be used in the high-pressure line, returning the condensed water to the boiler direct. The steam trap is usually a closed container provided with a small valve controlled by a float or bucket in such a manner that when the container becomes full of condensed water a small valve is opened and the steam pressure forces the condensate through the pipes to its destination. These small valves require an enormous amount of care to keep tight, otherwise the high-pressure steam blows through the trap and is a continual waste. Use as few traps as possible. The receiver with pump is a better device than the trap and is used in the better class installations.

Steam Loop.—A much better device than the trap for returning drips to the boiler is the steam loop of which there are a number of varieties. In general the system consists of a riser from the drip point, a length of uncovered pipe slightly sloping toward the discharge end and a drop leg connected to the mud drum of the boiler with a check stop valve and bleeder. The condensation and cooling in the uncovered horizontal pipe creates a sufficient drop in pressure to bring mixed steam and water to the upper level of the apparatus where it cools and by its weight forces itself past the check valve into the mud drum. The steam loop is somewhat modified in the larger systems but although successful when well installed is not as good a system as the receiver and pump.

Fig. 145.—Cochrane separator.

Steam Separators.—The separator is an enlargement of the piping system in which the steam velocity is reduced and the particles forced to travel in a zigzag direction against surfaces, corrugated or provided with lips to catch and retain the liquid particles which run down in the wake of the lips out of contact with the steam into a receiver. Some separators have screen baffle, others grids, but all embody the two principles, change of direction and reduction of velocity. Separators are built with either cast-iron, cast-steel, or riveted-steel shells but the heads are always castings.

Although absurdly high efficiencies are claimed by manufacturers of steam separators, efficiencies which cannot be realized in commercial practice, yet no separator should be retained in service that does not remove at least 80 per cent. of the water carried by the steam approaching the separator.

PROBLEMS

52. A 6-in. pipe line 150 ft. long carrying steam at 125 lb. gage pressure, was put up without expansion joints. At the end of the 150-ft. run the direction of the pipe was changed abruptly 90° and the pipe rigidly held in the new direction. Close to the 150-

ft. point several short ½-in. connections were tapped into the 6-in. main. As soon as steam was turned on the 6-in. ell was ruptured and every connection torn off. Why? How much did the pipe move at the point in question?

53. Determine the proper steam and exhaust pipe sizes for a 325-i.hp. non-condensing Corliss engine. What would be the pipe sizes if this engine were designed to run condensing?

What would be the sizes for a 6,000-hp. condensing engine?

54. A 100-hp. Corliss compound condensing engine requires the following steam per indicated horsepower-hour.

Percentage of load	25	50	75	100	125
Pounds steam per indicated horsepower-hour	32	24	21	20	20.5

If the steam pressure is 100 lb. gage, what size steam and exhaust pipes ought the engine to have? What is the velocity of the steam in the pipes for the various loads on the engine?

55. Calculate the velocities in the various pipes as given by Salmon's formula. Plot them with pressures as abscissæ. Deduce the equation to the curve. Plot the curve

$$Y = \frac{36,000}{X^{.25}}$$

and compare.

CHAPTER XI

COAL AND ASH HANDLING

Coal Handling.—For small plants coal is usually hauled by wagons and dumped in the boiler room in front of the boilers, but it is better practice, even in these small plants, to provide a pocket above the boilers

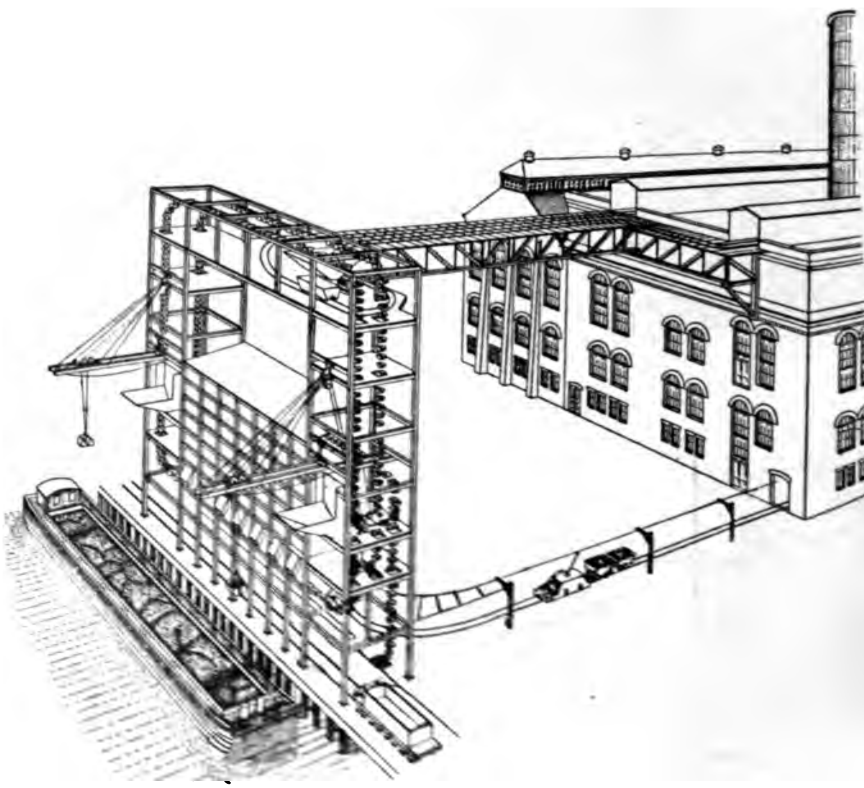


FIG. 146.—Coal handling machinery Williamsburg Power House Transit Development Co., Brooklyn, N. Y.

with some means for getting the coal into it. The coal may be dumped from the car or wagon into a small receiving hopper, which feeds a traveling belt, conveyor bucket or elevator system, which transfers the coal to the overhead pocket. In some stations the car is unloaded by means of a grab bucket on a telpher system which transfers the coal, a bucketfull at a time, from the car or wagon to the coal pocket.

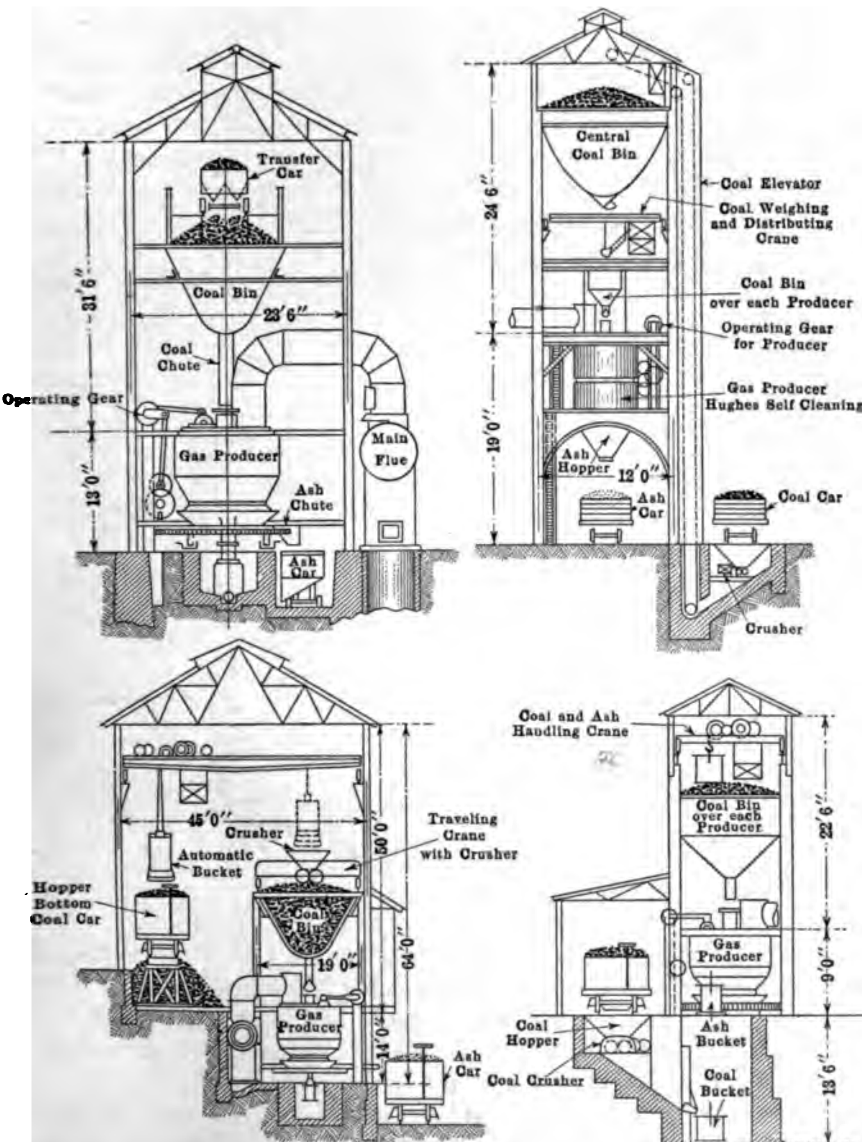


FIG. 147.—Typical coal-handling apparatus for gas producers or boilers.

Where it is not possible to install a coal pocket it is customary to have a storage pile in the neighborhood of the fire room. Industrial railway tracks are run from the pile to the boiler fronts and small cars may be loaded either by hand or by a small locomotive crane and run along the tracks in front of the boilers, the hand-firing being done direct from the cars.

This outside coal storage has been adopted for a number of larger stations, but small hoppers are provided above each boiler capable of holding 2 or 3 tons of coal and a car system, or telpher system, has been installed from the coal pile to the fire room to keep these hoppers full.

The design of coal-handling machinery depends largely on local conditions. In one station of 15,000 hp. the location is on a hillside, the coal supply being obtained by wagons which drive onto the roof of the

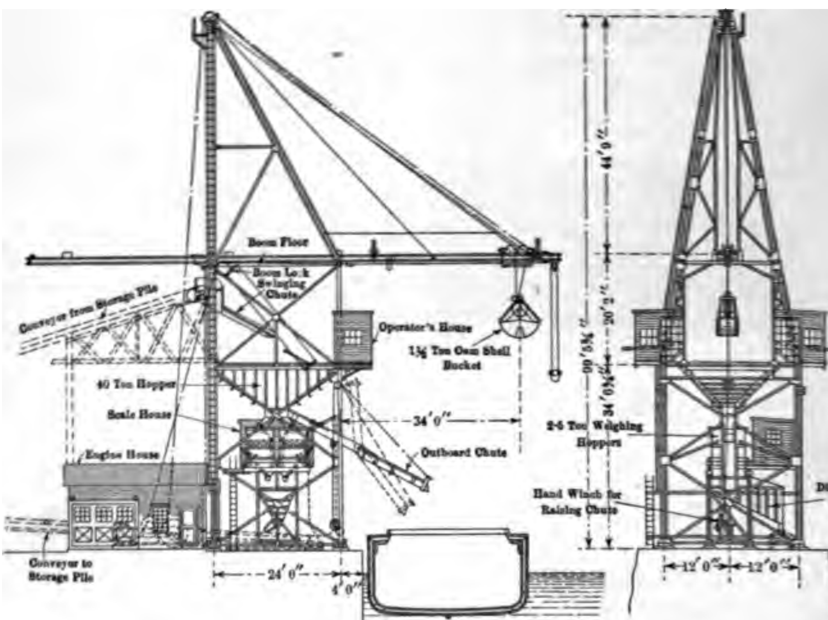


FIG. 148.—Typical coal-handling tower, one-man type.

station and dump their coal directly into the coal pocket. At Carville, near Newcastle, England, the railroad siding is extended over the roof of the boiler house and the coal is delivered direct from cars to pocket.

Most of the larger stations in the Eastern States receive their coal by water. The coal is removed from the boats by means of clamshell buckets, often as large as 2 tons capacity, which are hoisted at once to a maximum height required for delivering the coal. This amounts in some stations to a hoist of 140 ft. The coal is dumped from the clamshell bucket into a receiving pocket and then passed through a crusher which reduces the coal to the proper size for firing. From the crusher it goes into the cars of a cable railroad or conveyor, which delivers it to the various coal pockets from which large downtakes lead to each stoker or firing door. The cable railroad is usually the cheapest conveyor to install, maintain and operate. Next comes the traveling belt. The bucket conveyor is the most costly of the three.

Forty to 50 tons of coal may be unloaded per 8-hr. day by one man.

One man has fired as much as 16 tons of coal per watch of 8 hr.

Cost of Coal Handling.—In small plants the cost of coal handling may be as large as 40 to 45 cts. per ton fired. Where machinery is installed the cost drops with careful design and increase of size of plant to the neighborhood of 8 to 12 cts. per ton with good sized plants. For the larger of the central stations this cost should not exceed 4 cts. per ton. The cost of stacking in well-designed storage piles should not exceed 2 cts. per ton and the cost of reclaiming will be about the same. It should be remembered that in small plants a good man and one mule will handle a considerable amount of coal at an exceedingly cheap rate, and in most of the smaller plants it will not pay to install coal-handling machinery.

An idea of the saving to be secured by mechanically handling the coal may be obtained from the following figures:

A plant of 7,500 hp. in boilers was operated for some time without coal-handling machinery other than small hand cars which were loaded by hand from railway cars outside of the building, and which were then hauled up a straight incline to the boiler house, so that the fuel could be dumped in front of the furnaces.

The coal-handling machinery later introduced was so arranged that the coal was only handled by hand in shoveling it out of railway cars onto the conveying system.

	Wages	Tons burned	Cost per ton
Hand operation:			
16 firemen and 1 helper,	$981.80	4,292	$0.229
11 coal and ash men. Ash removed by contract	634.66	....	0.1478
Mechanical operation:			
3 firemen and 2 helpers,	287.75	6,975	0.041
11 coal and ash men, 2 conveyor men	654.50	....	0.0938

The saving in wages of firemen and helpers amounted to 18.8 cts. per ton, which is 82.1 per cent. or $1,311.30 per month.

The saving on coal and ash handling is 5.4 cts. per ton, which is 41.4 per cent. or $376.55 per month, or a total saving of $1,687.85 per month or over $20,000 per year.

If the coal did not have to be shoveled from coal cars onto the conveyor in this plant the saving on labor might be even greater.

The total cost of handling coal from coal car to ash car in large central stations is roughly 10 to 18 cts. per ton.

Letters from owners of about 600 boilers to Mr. R. S. Hale of the Steam Users' Association, indicate that it costs to move coal by hand (wheelbarrow) about 1.6 cts. per ton-yard up to distances of 5 yd., then about 0.1 ct. per ton-yard for each additional yard.

One man, besides a night man, can run an engine and fire up to about 10 tons of coal per week.

One man, besides an engineer and night man, can fire up to about 35 tons per week.

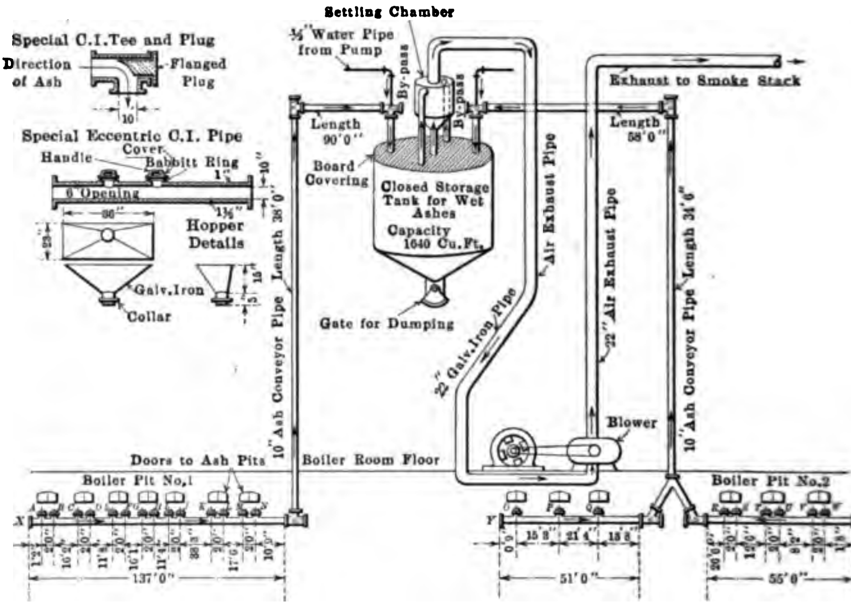


FIG. 149.—Diagrammatic view of pneumatic ash-conveyor plant, Armour Glue Works, Chicago, Ill.

Two men, besides an engineer and night man, can fire up to about 80 tons per week. These figures assume that the night man does all he can of the banking, cleaning and starting.

Mr. Hale further reports that mechanical stokers save 30 to 40 per cent. of labor in plants burning from 50 to 150 tons per week, and save no labor in small plants.

Boiler attendants were paid about $1.50 per day of 10 or 12 hr.

Average cost of firing coal was reported to be 48 cts. per ton; maximum, 71 cts. per ton; minimum, 26 cts. per ton.

Ash Handling.—In small hand-fired plants and in most of the older plants no ash pockets were provided under the grates, the firemen pulling the ashes out of the ashpit with a hoe and then shoveling them into ash cans, wheelbarrows or cars for removal. Where the power plant has a basement ash hoppers allowing the storage of about 8 hr. ashes are usually provided. These hoppers have suitable gates at the bottom, through which the ashes may be dumped into steel dump cars which are hauled by an electric locomotive outside the station. In the country the ashes are generally used for filling in adjoining land; in the interior cities they are usually valuable, in some cases bringing as high a price as 50 cts. per cubic yard. Where they cannot be used for these purposes the dump cars deliver into a skip or ash hoist which delivers the ashes into a pocket from which they may be delivered into boats, cars or wagons for disposal. At the Armour Glue Factory in Chicago is installed a pneumatic ash system, taking care of the ashes from 22 B. & W. boilers of 300 hp. each. In this installation an 8-in. cast-iron pipe is run along the front of the ash hoppers which are large enough for 24-hr. storage. The hopper is provided on an 8 by 12-in. aperture in the pipe and the ashes are pulled from the ash hopper into the pipe by hand. A vacuum of from 10 to 15 in. of water is maintained in the pipe by an exhauster of the Connersville double-impeller type placed at the outboard end of the 8-in. line. In front of the exhauster a separator is placed similar in principle to the standard shaving separator and discharged through a valve in the bottom into the ash pocket. With this system the ashes are conveyed horizontally 200 ft. around numerous bends, and vertically 70 ft. before being discharged into the hopper. The ashes must be dry and all large lumps and clinkers must be broken up. The operation is by one man and is said to take not over 2 hr. per day.

Traveling belts and bucket conveyors have also been used for handling ashes, but the upkeep on this apparatus is very large indeed.

Cost of Ash Handling.—The cost of getting the ashes out of the station and into the ash pocket will vary from 25 cts. per ton of coal fired in small stations to as low as 4 or 5 cts. per ton of coal fired in very large stations with well-designed apparatus. The cost of removing ashes varies with the locality and may be a source of revenue under certain conditions. No figures can be given for the maintenance of ash-handling plants, but as a general rule, the maintenance will be at least as large as the labor cost, as ash-handling machinery wears out rapidly. The cost of ash handling varies also with the kind and quality of coal, being larger as the percentage of ash increases, and much larger with clinkering coal.

Coal Storage.—It is usual in power stations to provide coal pocket space for from 7 to 10 days' supply of coal. This, however, is insufficient for a central station and further storage must be provided outside of the station. This usually consists of a coal pile, ranging from 10,000 to 200,000 tons capacity, depending on the size of the central-station system, it usually being considered that from 3 to 6 months' supply is sufficient to insure continuity of operation under all circumstances. These piles are provided with stacking and loading machinery and are of two general types, one in which the pile is spanned by a bridge carrying the loading and unloading machinery, and the second type in which the pile is made by means of trestles and cable cars, while the unloading is done by means of locomotive cranes and the cable-car system. A third type, largely used by the hard-coal railroads, consists of conical piles about 400 ft. in diameter and 90 ft. high, in which the small sizes of anthracite coal are taken to the top of the pile by means of inclined flight conveyors, and the reclaiming is done by a movable horizontal flight conveyor.

CHAPTER XII

THE STEAM POWER PLANT

OPERATING COST

Design of Power Plant.—No better service can be rendered the non-expert about to construct a power plant than to advise him to engage a capable engineer to design the plant and to superintend its installation.

Methods of Buying Apparatus.—

(*a*) Bids.

(*b*) Have reputable manufacturer build it and pay what he asks.

(*c*) Have engineer state in specifications requirements of apparatus wanted, permitting manufacturer to vary details enough to enable him to use standard designs.

(*d*) Have engineer design the whole plant in detail buying standard apparatus where possible but developing new designs to meet new conditions.

Location.—The most important factors governing the location of the power plant are:

(*a*) Availability of water supply, especially for condensing.

(*b*) Economical handling of fuel and ash.

(*c*) Storage capacity for fuel.

(*d*) Ease of power distribution.

(*a*) The supply of water must be guaranteed and must be abundant. Wells do not usually furnish desirable water for boiler purposes, lake or river water being preferable. Ferranti states:

> "The water supply is enormously important today and I see no method so long as the converting process is thermal, that is to say, where there is a rise and a reduction in temperature, where the cooling water will not play a most important part."

(*b*) If possible, the boiler house should be so located that coal can be delivered directly from boats or cars to the storage bins. If the grade can be so arranged as to avoid elevating the coal, a saving of from 25 to 50 cts. per ton may often be effected.

(*c*) Ample coal storage capacity should be provided to serve in times of strikes, blockages, etc.

(*d*) This depends upon the character and purpose of the plant. Provisions should always be made for future enlargements and extensions.

Constructions.—The type of building is determined by the price of land and the available space. The engine and boiler house should be separate from the other buildings, to avoid danger from fire and to prevent troubles from vibration.

Where the ground is not too expensive the entire plant may be on one level. In this case a one-story building with brick walls, steel-truss roof, and cement floor is most common. A well-drained and well-lighted basement should be provided if the plant is large for pipes, condensers, ash hoppers, traps and such apparatus as is necessary below grade. In large plants basements are often 20 to 25 ft. in the clear.

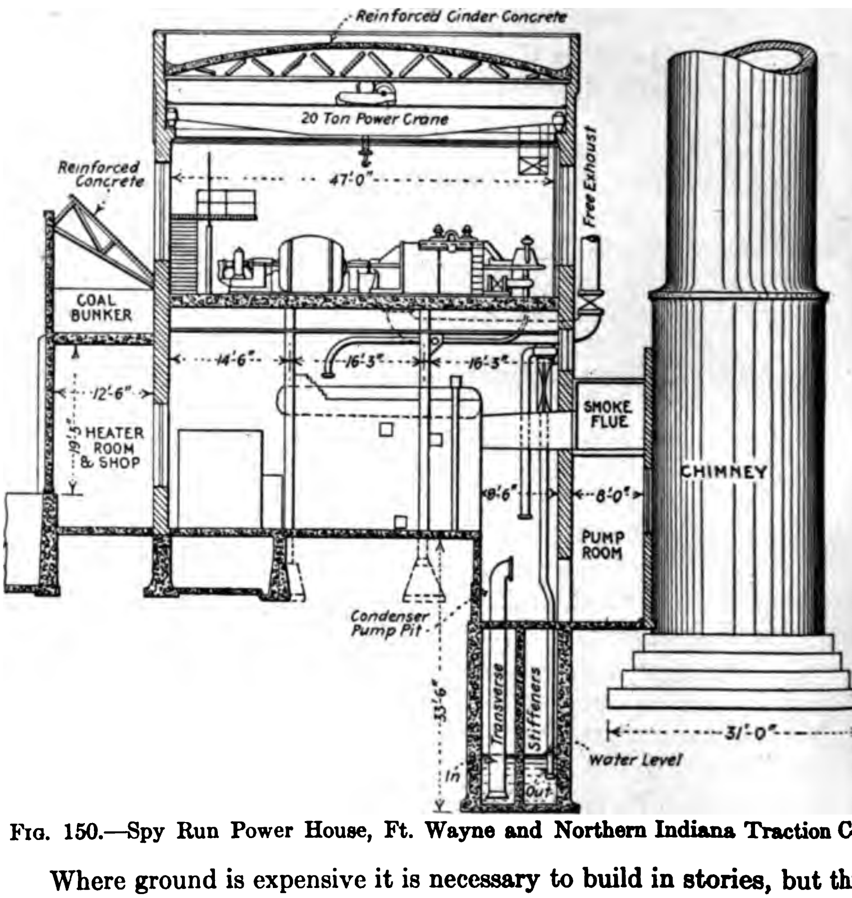


FIG. 150.—Spy Run Power House, Ft. Wayne and Northern Indiana Traction Co.

Where ground is expensive it is necessary to build in stories, but this practice should be avoided when possible.

It is a question whether engines or boilers should be placed above. Formerly the boilers were placed above the engines, but of late the plan has been reversed.

All pumps, heaters, etc., should be on the main floor where practicable for the sake of light and cleanliness. The engines and boilers should be in separate rooms with a tight partition between to keep dust and

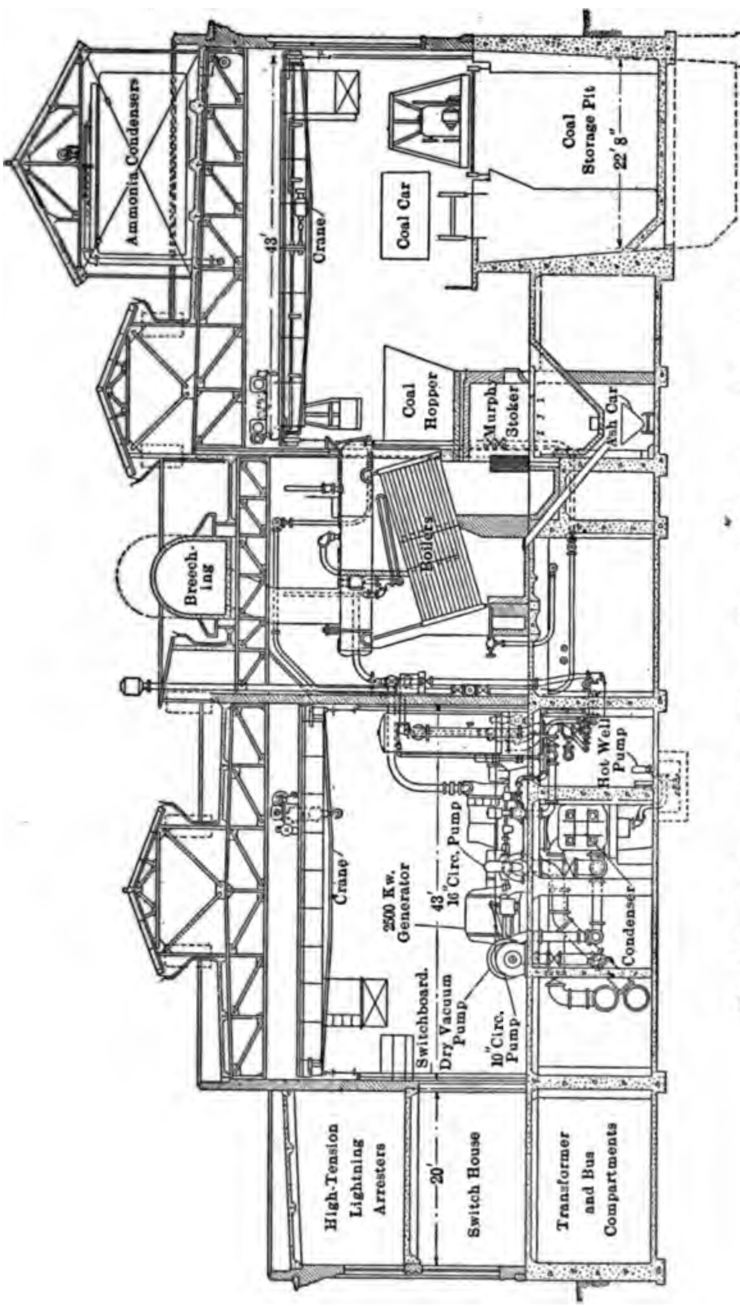


FIG. 151.—Lexington Power Station, Kentucky Traction and Terminal Co.

[illegible]

[illegible] should be [illegible]

[illegible]

[illegible]

[illegible] pumps with all the [illegible] plants to have an injector connected di-[illegible] emergencies but this is too expensive for [illegible] provision must be made for filling the boilers [illegible] city mains or by hand pumps.

[illegible] requirements of station systems are reliability, [illegible] and durability. The engineer is concerned in the design of [illegible] require a shutdown of the entire system. The [illegible] must determine the system. If the service is 24 hr. [illegible] at half load only, then the system should provide [illegible] at the same time maintaining the capacity. If [illegible] required to generate at all times two-thirds of its total capac-[illegible] the plant should not have less than three units all of the same size. [illegible] should be so large that the station cannot be operated without [illegible] the other machinery to the permissible limit.

[illegible] and economy in large central stations all units should be of the same size and the number of generating units should not be less than [illegible] nor more than eight, one unit in reserve, the others for carrying the load.

In any case the original installation should be for a two-division, three-division or four-division plant with provision for future divisions. If the plant is to be a two-division plant it may be built with two units, each of a capacity slightly larger than the minimum load conditions demand, and the future unit may be made twice the capacity of these smaller ones.

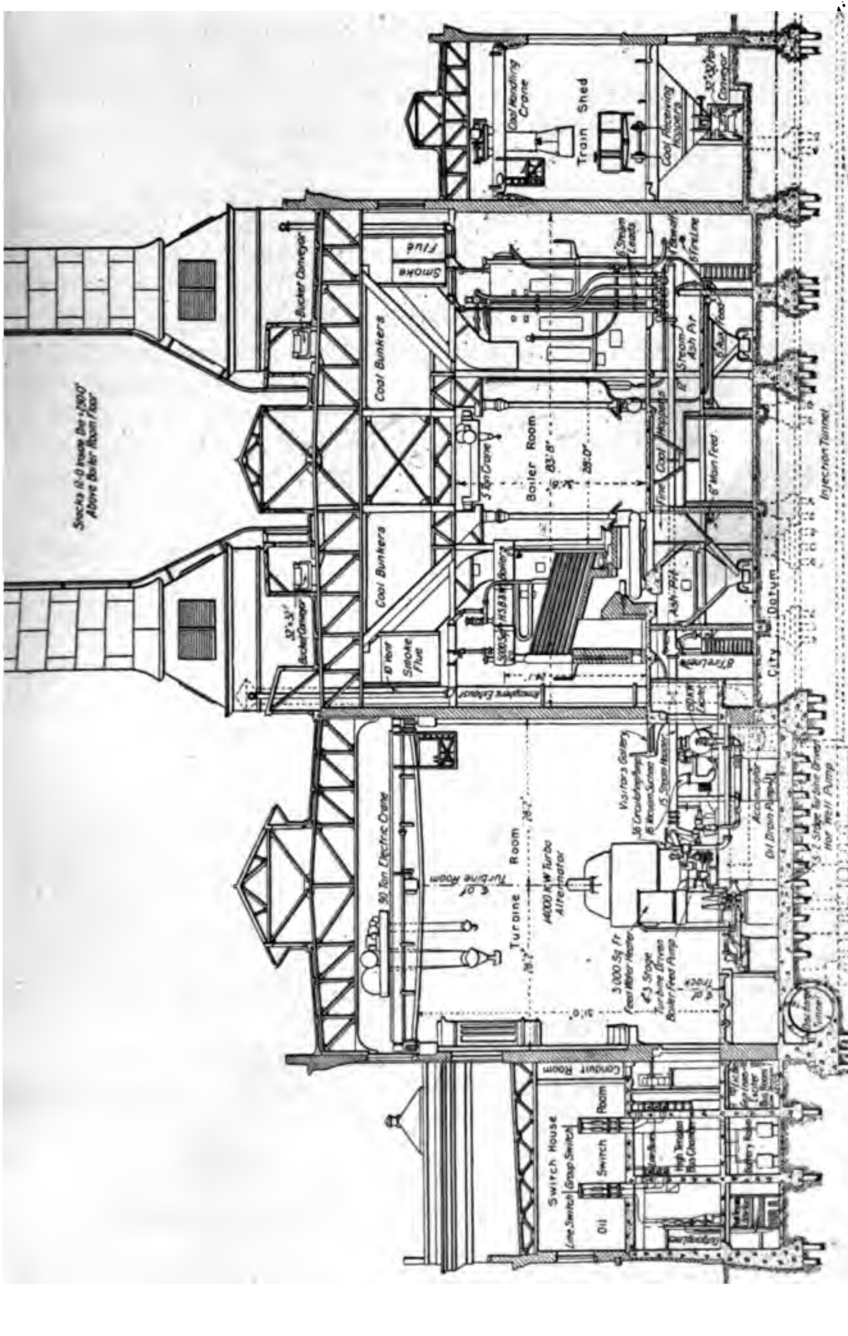


FIG. 152.—Cross-section of Quarry St. Station, Commonwealth Edison Co., Chicago, Ill.

This arrangement of units would be permissible in a plant which ordinarily delivers only half of the power called for on Sundays, holidays, etc. It has been found ordinarily that a three-division plant is more suitable for permitting repairs to be made and also requires less investment for reserve capacity. If the plant is to be a three-division plant, the boilers should be in three divisions, say, 6, 9, or 12 in number.

The auxiliaries should be in pairs, not necessarily together, but each sufficient for the entire plant, and they should be connected to the different divisions so that when one division is out of service the other auxiliary will be on operating division.

Let it be assumed that a station requires three units for an output of two-thirds of the total capacity of the units and later will require an additional unit. There is but one solution for the problem under these conditions and that is that the station shall ultimately be a four-division plant for output of three-quarters of the total capacity of the units. The boilers must be arranged in three units so as to allow one to be out of condition for cleaning and repairs.

Assuming the engines to be of 2,000 hp. each, then two 500-hp. boilers would be required for each engine, but by assuming three engines of 2,000 hp. each or a total of 6,000 hp. there would be 3,000 hp. in boilers to be divided into five units, so as to allow one boiler to be out of service. This would give five 600-hp. boilers. If the fourth engine unit is likely to be ordered before the three units are called upon to carry full load for a large part of the time it would be safe to estimate on 8,000 hp. of engines or 4,000 hp. of boilers, or seven boilers, each of 555 hp., which is a somewhat better arrangement for the four-unit plant.

Types of Station Design.—In the United States, where fuel has as a rule been reasonably cheap, the standard type of power station includes the boiler and engine with steam-driven auxiliaries. Economizers are very rarely used and the auxiliaries are run non-condensing, the exhaust steam being led to a heater and used for heating the feed water. This may be termed standard American design, and exceedingly good results may be obtained from it.

In Europe, where coal is as a rule poorer and more costly than in America, and where interest charges have been considerably less, another type of station is more usual. Economizers are the rule and usually at least one-half as much surface is furnished in the economizer as in the boilers. The auxiliaries are driven direct from the prime mover or by electric motors, and no feed-water heaters are used because of the lack of available exhaust steam. The condensate is used over again, being pumped directly into the economizers. Such stations, when well designed, are capable of exceedingly good economy, but apparently no better than is given by the American type of plant.

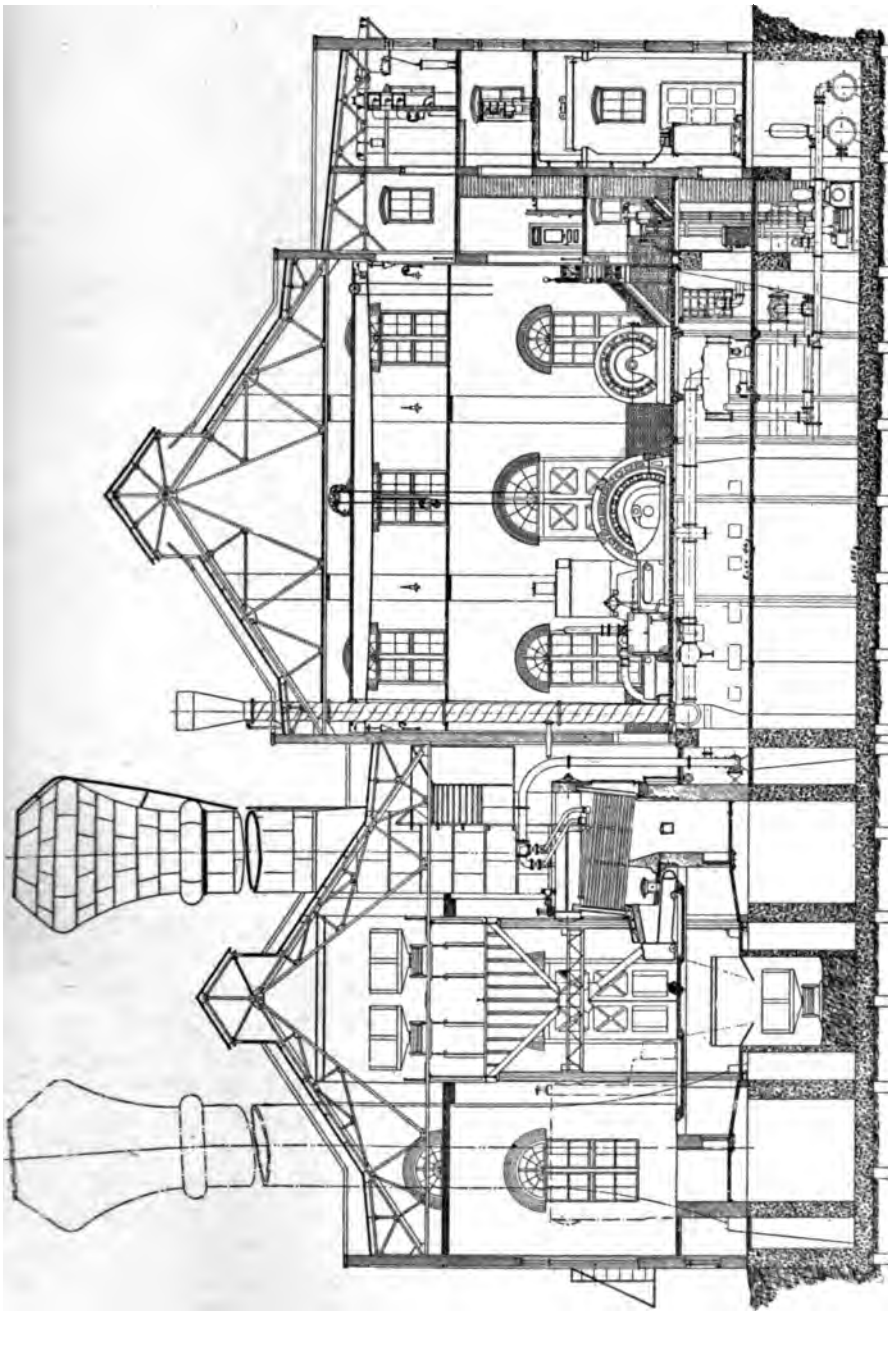


FIG. 153.—Cross-section of Illinois Traction Co.'s station.

There are a number of American plants following to some extent the European types, but differing from them widely in the types of apparatus. In these plants both economizers and feed-water heaters are present, together with a number of other heat-saving devices. The auxiliaries are electrically driven from current supplied by a separate auxiliary unit, which has a jet condenser, in which the feed water of the station is used as the condensing water. There are many modifications of this type possible, and notable examples exist in the Conners Creek station of the Detroit Edison Co. and the Cleveland Municipal Plant. It is doubtful whether better results under similar conditions will be obtained by this type of station.

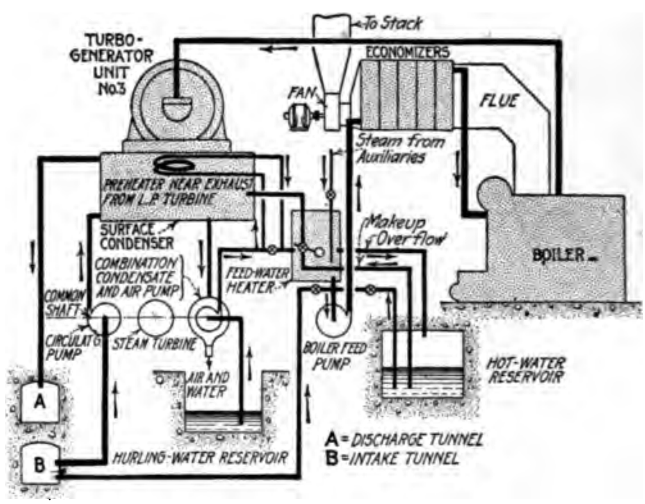


FIG. 154.—Diagram of steam and water circuits Northwest Station, Commonwealth Edison Co., Chicago, Ill.

These three types may well be represented by the apices of an equilateral triangle; between them lies the whole field of the many variations which occur in design. Given the same load factor and use factor and with equally careful design, it is probable that similar economic results may be obtained from any type of plant, but to secure these results the design must be of the very best and the local conditions must be good. In particular the design must be suited to the local conditions and the type of fuel procurable in that locality.

A fourth type of power station, which perhaps under modern conditions might be as economical as any, is the power plant of the steamer Inch Dene, a 10-knot freighter using Scotch marine boilers, superheated steam, multiple expansion engines, with the auxiliaries driven from the engine itself. The economical results from these ships have been exceedingly good, but it should be remembered that the use factor in this case is practically 90 per cent. Particulars of a test of this vessel may be found in *Marine Engineering*, vol. 6, p. 332.

At the present time there is renewed agitation for the use of higher pressure steam, and a number of manufacturers are ready to supply boilers designed to furnish steam at 600 lb. pressure and 200° superheat. The turbine manufacturers are prepared to offer turbines to utilize this high-pressure steam. Whether these changed conditions will result in a new type of station cannot be stated, but it is certain that additional operating economies may be secured, although the total economy, when fixed charges are considered, may not be much better.

It is not probable that any one type of station will become standard, but that the development of the three types with their variations will proceed along similar lines until a change takes place in our method of generating power.

Cost of Buildings.—The estimated cost of buildings is most readily determined on the basis of cost per square foot of floor space or cost per cubic foot of space for the entire building.

The following figures represent the averages of several quotations:

	Cost per square foot, $	Cost per cubic foot, cents
Mill construction	0.80–1.10	6.5– 8.5
Fireproof stores, factories and warehouses with brick, concrete, stone and steel construction	2.00–3.00	14.0–25.0
Concrete or reinforced-concrete shops, factories and warehouses	1.25–1.75	8.0–16.0
Plain power houses with concrete floors and with brick and steel superstructure	2.00–2.75	9.0–12.0
Power houses under city conditions with superior architectural details	3.00–4.50	15.0–30.0

A consulting engineer of large experience finds the cost of boiler houses, engine houses and coal pockets to run approximately as follows when based on the cost per engine horsepower installed.

Cost of Buildings per Engine Horsepower

Simple non-condensing engines:									
Engine horsepower	10	12	14	15	20	30	40	50	75
Boiler house, cost per hp.	$37.15	$33.00	$30.00	$28.50	$24.50	$20.50	$18.00	$16.00	$13.00
Engine house, cost per hp.	4.80	4.35	4.00	3.90	3.30	2.75	2.50	2.30	2.15
Coal pocket, cost per hp.	20.00	18.00	16.00	15.00	13.70	11.00	9.80	8.30	6.00
Simple condensing engines:									
Engine horsepower	10	12	14	15	20	30			
Boiler house, cost per hp.	$33.70	$29.60	$27.50	$26.20	$21.60	$18.20			
Engine house, cost per hp.	14.40	12.60	11.30	10.90	8.60	7.75			
Coal pocket, cost per hp.	19.00	17.90	16.60	15.80	13.60	11.00			

Cost of Buildings per Engine Horsepower.—(*Continued*)

Engine horsepower	40	50	75	100		
Boiler house, cost per hp.	$16.00	$14.80	$11.30	$9.70		
Engine house, cost per hp.	6.40	5.35	4.90	4.30		
Coal pocket, cost per hp.	8.70	8.50	6.30	5.70		
Compound condensing engines:						
Engine horsepower	100	200	300	400	500	600
Boiler house, cost per hp. } Engine house, cost per hp. }	$28.50	$24.00	$11.20	$8.00	$6.40	$5.70
			11.20	9.35	8.50	7.20
Coal pocket, cost per hp.	5.70	4.00	3.10	2.60	2.40	2.25
Engine horsepower	700	800	900	1,000	1,500	2,000
Boiler house, cost per hp.	$5.35	$5.00	$4.70	$4.55	$4.10	$3.95
Engine house, cost per hp.	6.30	5.60	5.35	5.00	4.75	4.55
Coal pocket, cost per hp.	2.10	2.05	1.95	1.80	1.75	1.60

Incidentals.—In erecting a power plant, there are always a lot of miscellaneous items that add to the total expense but which are very difficult to determine before the installation is made. These always amount to considerably more than anticipated, usually averaging in the neighborhood of 10 to 15 per cent. of the cost of the project.

If based on the engine horsepower, the incidentals reported by one consulting engineer run about as follows:

Cost of Incidentals per Engine Horsepower

Simple non-condensing:						
Engine horsepower	10	12	14	15	20	
Incidentals, cost per horsepower	$20.00	$18.00	$16.00	$15.00	$13.70	
Engine horsepower	30	40	50	75		
Incidentals, cost per horsepower	$11.00	$9.80	$8.30	$6.00		
Simple condensing:						
Engine horsepower	10	12	14	15	20	30
Incidentals, cost per horsepower	$18.00	$17.60	$17.00	$16.60	$14.70	$11.80
Engine horsepower	40	50	75	100		
Incidentals, cost per horsepower	$10.80	$10.30	$8.70	$7.80		
Compound condensing:						
Engine horsepower	100	200	300	400	500	600
Incidentals, cost per horsepower	$9.70	$8.00	$6.80	$6.50	$6.25	$6.00
Engine horsepower	700	800	900	1,000	1,500	2,000
Incidentals, cost per horsepower	$5.75	$5.50	$5.25	$5.00	$4.75	$4.60

Cost of Installations Complete.—An idea of the approximate cost of complete installations may be had from the following table.

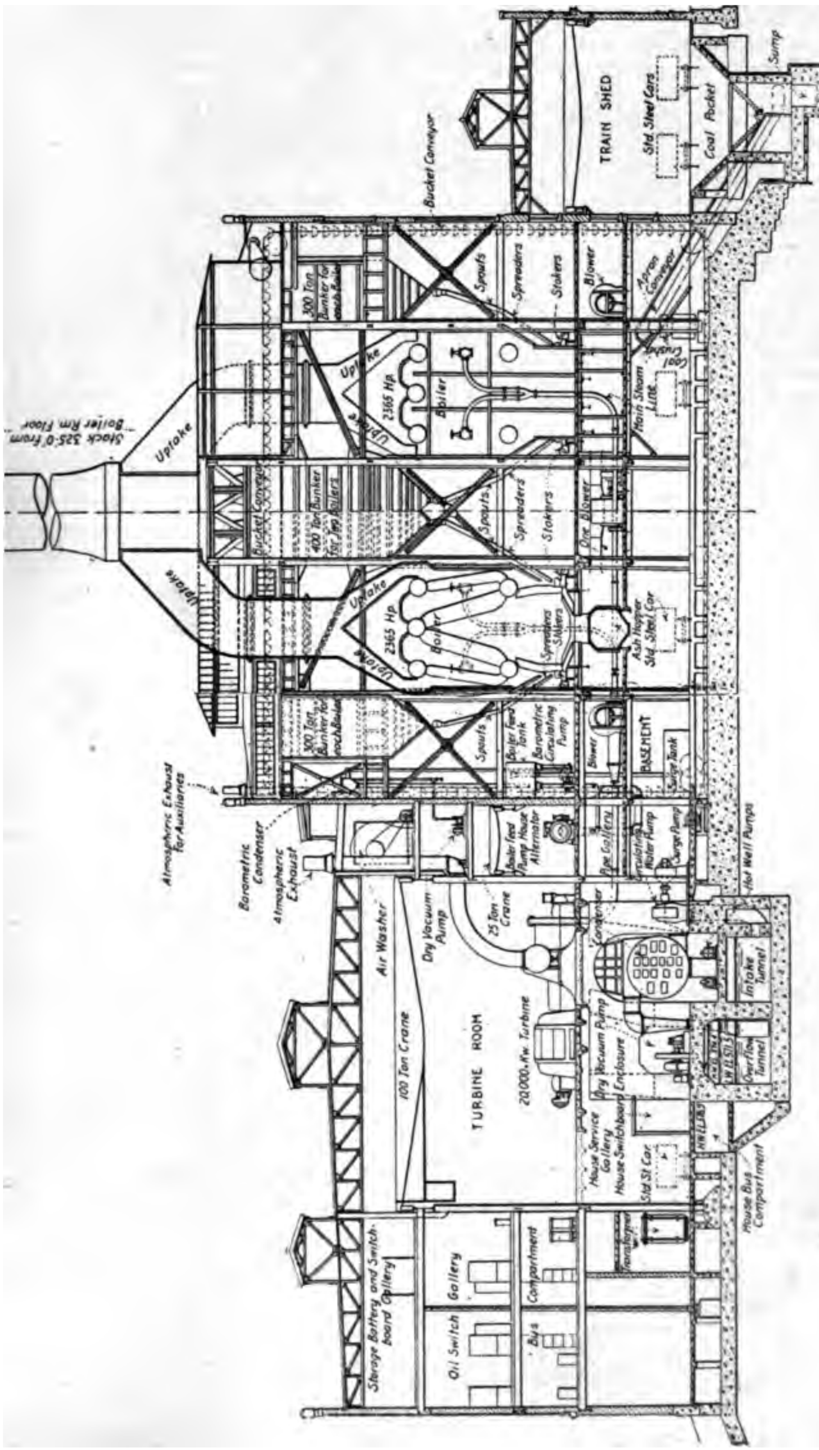


FIG. 155.—Cross-section Conners Creek Station, Detroit Edison Co.

APPROXIMATE COST PER KILOWATT OF STEAM TURBINE-DRIVEN INSTALLATIONS

	Size of plants—kilowatts					
	5,000	10,000	20,000	30,000	40,000	50,000
Building, real estate and excavating	$17.50	$14.00	$13.10	$11.65	$10.95	$10.30
Turbines and generators	28.25	23.50	21.20	18.75	17.65	16.50
Condensers	6.85	5.70	5.15	4.50	4.30	4.00
Boilers, stokers, superheaters and stacks	34.50	28.70	25.80	23.00	21.50	20.00
Bunkers, and conveyors	5.75	4.80	4.30	3.85	3.80	3.40
Boiler feed and service pumps	1.20	1.00	0.90	0.80	0.75	0.70
Feed-water heaters	1.90	1.60	1.45	1.30	1.20	1.10
Switchboard and wiring	4.20	3.50	3.15	2.80	2.60	2.50
Exciters	2.50	2.10	1.90	1.70	1.60	1.50
Foundation (machinery)	1.30	1.10	1.00	0.90	0.85	0.75
Piping and conduits	8.50	7.10	6.40	5.70	5.30	5.00
Crane	1.80	1.50	1.35	1.20	1.10	1.00
Supt. and engineering, etc.	5.75	4.80	4.30	3.85	3.60	3.35
	$120.00	$100.00	$90.00	$80.00	$75.00	$70.00

Averaging the figures above shows the percentage distribution of cost to be approximately:

	Per cent. of total cost
Building, real estate and excavating	14.6
Turbines and generators	23.5
Condensers	5.7
Boilers, stokers, superheaters and stacks	28.7
Bunkers and conveyors	4.8
Boiler feed and service pumps	1.0
Feed-water heaters	1.6
Switchboard and wiring	3.5
Exciters	2.1
Foundation (machinery)	1.1
Piping and conduits	7.1
Crane	1.5
Supt. and engineering, etc.	4.8
	100.00

Comparative Cost of Steam Power Stations, Complete.—The following values will illustrate the relative cost of different types of power stations. The figures are for complete plants including engines, generators, boilers, piping, feed pumps and heaters, stacks and buildings. Direct-connected generators—one reserve unit.

COST OF STEAM POWER STATIONS

Horsepower	100	200	400	600	1,200	2,000
Simple non-condensing high-speed	$12,960	$19,280	$32,210	$43,950	$78,300	
Compound non-condensing high-speed	14,140	21,180	34,370	45,940	79,860	
Compound condensing	15,230	22,740	37,400	49,890	85,990	
DeLaval turbine	14,860	21,180	36,050	48,300	84,240	
Vertical condensing low-speed	......	......	52,280	69,470	108,990	155,930
Horizontal condensing low-speed	......	......	51,420	69,490	102,500	148,310
Parsons turbine	.....	......	......	60,060	88,090	122,100

COST PER KILOWATT OF SEVERAL STATIONS

	Electric Ry. Economy, 1903, McGraw Pub. Co.		Yorkshire Power Co., 6,000 kw. Thornhill	10,000-kw. engine plant	90,000-kw. turbine plant	150,000-kw. plant (London)	Greenwich London, 34,000 kw.
	Max.	Min.					
Land	......	.....	$0.43	......	$2.14	$1.60	
Foundations	$3.50	$1.50	.....	$1.22	4.17		
Buildings	15.00	8.00	.....	9.70		14.50	
Stacks	2.00	1.00	.....	1.22			
Total building	20.50	10.50	13.4	12.14		16.10	$56.55
Boilers	17.00	9.00	.....	12.10			
Superheaters	......	.....	.....	......			
Stokers	3.00	2.50	.....	2.42	17.80	6.45	14.69
Economizers	4.50	2.50	.....	1.46			
Coal and ash	7.50	3.00	.....	1.94		1.94	1.56
Piping	12.00	4.00	.....	7.30		3.40	7.86
Heaters and pumps	3.00	2.00	.....	0.97			
Prime movers	53.00	38.00	.....	58.30		13.85	28.40
Condensers	......	.....	.....	6.30	10.55		
Crane	......	.....	.....	......			1.07
Exciters	......	.....	.....	1.46		2.13	
Switchboard cables	10.00	4.50	.....	2.42	2.67		5.12
Incidentals	2.00	2.00	.....	......			
Total equipment	112.00	67.50	78.00	94.67		27.77	58.70
Total	132.50	78.00	91.40	106.81	37.33	43.87	115.25

STEVENS and HOBART SNELL, 1912. RIDER, I. E. E., 1909.

Operating Expenses.—Before deciding on the best and most economical plant for a given set of conditions, it is necessary to consider the relative yearly expense. This may be divided into fixed charges which are a function of the cost of the plant and must be paid whether or not power is produced; and operating cost or as it is often termed, station cost. The original division by Hopkinson in 1892 was into two similar categories: (*a*) a fixed charge depending on the maximum rate at which the energy may be demanded and independent of the time over which the demand may extend; and (*b*) a running charge proportional to the time the demand is kept up. His fixed charge might be termed a "readiness to serve charge" and included interest on cost, taxes, insurance, amortization plus the expense for labor, fuel, stores, etc., needed to keep the plant running light and ready to work. His running charge[1] included the additional fuel, labor, repairs, and stores necessary for the carrying of the load.

Hopkinson's categories proved to be subject to disadvantages due to the difficulty of segregating "light running charges" from the other running charges but the substantial accuracy of the method has never been

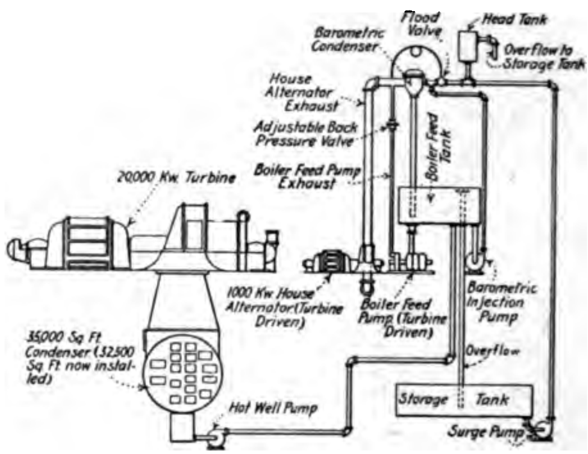


FIG. 156.—Diagram of auxiliary connections Conners Creek Station, Detroit Edison Co.

denied. He gave figures for a 2,500-kw. plant in which the "running light" charges amounted to $136,000 and the fully loaded charges to $288,000. The "light running" charge per kilowatt of demand is not far from $54.30 and the corresponding costs of production including standing and running charges for the various use factors are given in the following table.

Per cent. use factor	Cost, cents	Per cent. use factor	Cost, cents
5	12.72	55	1.74
10	6.68	60	1.66
15	4.66	65	1.58
20	3.66	70	1.52
25	3.06	75	1.46
30	2.66	80	1.40
35	2.38	85	1.36
40	2.16	90	1.32
45	1.98	95	1.28
50	1.86	100	1.26

[1] A very good account of the discussion of these principles may be found in WORDINGHAM'S "Central Electrical Stations" and in the *Transactions* of the Institute of Electrical Engineers (London, Eng.).

The later and present practice is to include interest, depreciation, taxes and insurance in fixed charges and fuel, labor, including superintendence, water, oil, waste and supplies, and maintenance in station cost or operating cost. The sum of the two is called total cost or better, production cost.

These may be tabulate as follows:

Fixed Charges:

(*a*) Interest on investment.
(*b*) Depreciation (replacement).
(*c*) Taxes, (city, state, etc.).
(*d*) Insurance.

Station Charges:

(*a*) Fuel.
(*b*) Labor (including superintendence).
(*c*) Oil, waste and supplies.
(*d*) Water.
(*e*) Maintenance.

Interest.—It is usually fair to allow 6 per cent. for interest on investment but small industrial plants usually have to pay higher rates, say 7 or 8 per cent. Municipalities may pay as low as 4 per cent. and it has been possible in Europe in the past to figure on a smaller return. For the general run of problems it is safe to use 6 per cent.

Depreciation.—Structures and machinery grow old and unfit for the purpose for which they were erected or purchased. They may be kept in reasonable repair by the ordinary running maintenance but there will come a time when the plant is worn out and must be replaced if the business is to continue. If the business ends with the life of the plant, as in some mining propositions, the capital has been destroyed and the investor only gets the interest return which in this particular case must be high. To meet the above conditions, it has grown to be the custom to set aside every year a sum of money which is known by various names, depending on how it enters into the accounts, such as sinking fund, depreciation or amortization. It may be paid to the stockholders either as extra dividends or by putting it into improving the value of the plant and these two ways are considered the best methods for private business and small close corporations. For public corporation plants the sinking-fund method is the best as in these plants the capital is furnished by the sale of bonds which must be redeemed at a certain time. All corporations other than close corporations should use the depreciation method and invest the depreciation fund so that when the old plant wears out or is superseded the money will be available to build a new one.

The very rapid development of the power generation and distribution business in the last 30 years has shown the difference between the actual life of power-generating machinery and its useful life; or its life up to the time that its use is superseded by larger and more economical apparatus or a machine better suited to changed conditions of power generation. The following table shows the actual life which has been estimated for various portions of steam power-plant equipment.

APPROXIMATE USEFUL LIFE OF VARIOUS PORTIONS OF STEAM POWER-PLANT EQUIPMENT

	Years
Buildings, brick or concrete	50
Buildings, wooden or sheet-iron	15
Chimneys, brick	50
Chimneys, self-sustaining steel	25 to 40
Chimneys, guyed sheet-iron	5 to 10
Boilers, water-tube	30 to 50
Boilers, fire-tube	15
Engines, slow-speed	25
Engines, high-speed	15
Turbines	10 to 20
Generators, direct-current	5 to 20
Generators, alternating-current	5 to 20
Motors	10 to 20
Pumps	25
Condensers, jet	10 to 20
Condensers, surface	10 to 20
Heaters, open	30
Heaters, closed	20
Economizers	5 to 10
Wiring	20
Belts	7
Coal conveyor, bucket	5 to 10
Coal conveyor, belt	2 to 5
Transformers, stationary	30
Rotary converters	25
Storage batteries	3 to 5
Piping, ordinary	12
Piping, first-class	20 to 30

So much depends upon the design and the conditions of operation that no fixed values can be definitely assigned and the above figures should be used with caution. Practice shows that most power-plant appliances become obsolete long before the limit of their useful life is reached.

The Traction Valuation Commission in Chicago in 1906 gave the following percentages for plant depreciation:

	Per cent.
Engines, Corliss, low-speed	3 to 5
Engines, automatic, high-speed	5 to 10
Cable-winding machinery	3

	Per cent.
Generators, direct-connected, modern	5
Generators, belted (depending on date)	5 to 10
Traveling cranes	2
Switchboard and all wiring	2
Piping	35
Pumps	5
Heaters, closed	6 to 10
Heaters, opened, if cast iron only	3
Breeching and connections, brick	5
Breeching and connections, steel	10
Boilers and settings, horizontal tubular	10
Boilers and settings, water-tube	3.5
Grates	10
Coal-handling machinery	6
Ash-handling machinery	8
Combined coal- and ash-handling machinery	7
Storage bins, steel	3 to 10
Miscellaneous items	5

"The above annual rates of depreciation have been used as a basis in ›reciating the power-plant equipment. Apparatus has been depreci- d at these rates down to 20 per cent. of the wearing value, the wearing ue being determined by subtracting the scrap value from the cost new. power-plant equipment has been considered as worth 20 per cent. of wearing value as long as it is in operating condition. Depreciation pplied to wearing value, as the apparatus is always worth scrap value."

The above percentages applied to a particular plant of 2,900-kw. ›acity give an approximate depreciation for the whole plant of 4 per t.

It is idle to attempt to figure actual depreciation on a power plant m the above figures as many extraneous conditions enter into the prob- ı. The difference between good and bad feed water might vary the per cent. allowed for boilers from 4 to 20 per cent.

For a well-maintained plant the allowance might be only about one- f as much for depreciation as for one poorly maintained. For design l comparison purposes it is best to assume a fixed percentage for de- ciation. It is customary to use 6 per cent. and the error from the of this figure is not likely to be large in the present state of the art.

Taxes and Insurance.—Taxes will vary from 1 to 2 per cent. of the ıe of the property. Insurance of buildings and machinery will vary ı 0.5 to 1.5 per cent. These two items are usually combined for nating purposes at 2 per cent.

TOTAL FIXED CHARGES.

'ixed charges for estimating purposes may then be taken as:
ıterest, 6 per cent.

Depreciation, 6 per cent.

Insurance and taxes, 2 per cent.

Total, 14 per cent.

This value may be used in estimates and for solving the problems in these notes.

This total allowance of 14 per cent. for fixed charges may be regarded as fair when the operating portions of such installations are alone considered.

When the buildings of brick or concrete make up a large proportion of the investment, an average of 11 per cent. for fixed charges is perhaps a better figure.

STATION COST.

Fuel.—The available fuels for power-station purposes are the steam sizes of anthracite coal, bituminous coal, oil and natural gas. The steam sizes of anthracite include all the finer grades from pea coal down to the refuse or culm. Culm which may contain as high as 35 per cent. ash costs about 25 cts. a ton at the mine, No. 3 buckwheat or rice coal about 50 cts., No. 1 buckwheat about 80 cts. and pea coal about $1. Bituminous coal, which varies greatly in quality, also varies much in price at the mines averaging from 90 cts. to $1.50 per ton. The average freight rate on coal is 1¼ cts. per ton-mile.

Anthracite coal is used in the anthracite regions and to a considerable extent in the regions round about extending to New York and Philadelphia, but the steam fuel of the whole country as a rule is bituminous coal. The price at the power station will vary from $12 in certain unaccessible localities down to 90 cts. near the mines.

Oil, usually the crude oil of commerce, varies in price per barrel of 42 gal. at the well from 40 to 60 cts.

Oil is handled by pipe lines in the regions near the wells but a good deal of it is water-borne and the freight is relatively much lower than for coal. Contracts for Mexican and Texas oil have been offered in New York and Philadelphia for $1.25 per barrel.

Oil will probably be an available fuel only in the Pacific States where coal is high and poor and in Texas and Oklahoma.

At a distance from the wells the price usually runs from 2 to 4 cts. per gallon.

Although natural gas is found in limited quantities in many sections of the United States, its use for power-plant purposes is largely in the region of Pennsylvania, Ohio and West Virginia. The gas is piped from the wells and costs from 10 to 30 cts. per 1,000 cu. ft.

The cost of fuel must always be ascertained for the particular locality as large variations in price occur in an unexpected way due to local conditions. Before the discovery of the Californian oil fields most of the steam fuel used in San Francisco was Welsh coal brought out as ballast or to help make up a cargo. For the same reason Canadian coal was largely used in Westphalia, Germany, before the outbreak of the European war.

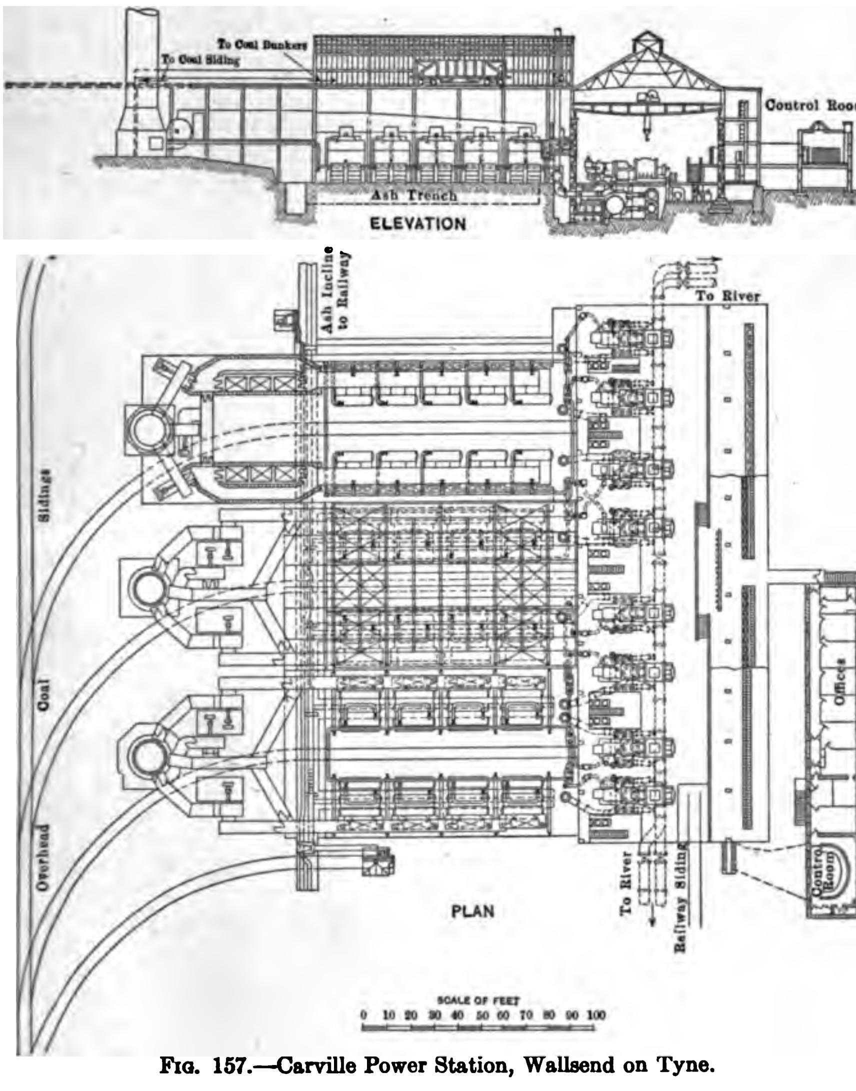


FIG. 157.—Carville Power Station, Wallsend on Tyne.

The price of the coal is not the only fuel cost. The coal must be put into the station bunkers, the ashes must be removed and disposed of and these costs should be added to the cost of coal. If the coal is insured, this cost is fuel cost and if stored for any time the interest cost should be added.

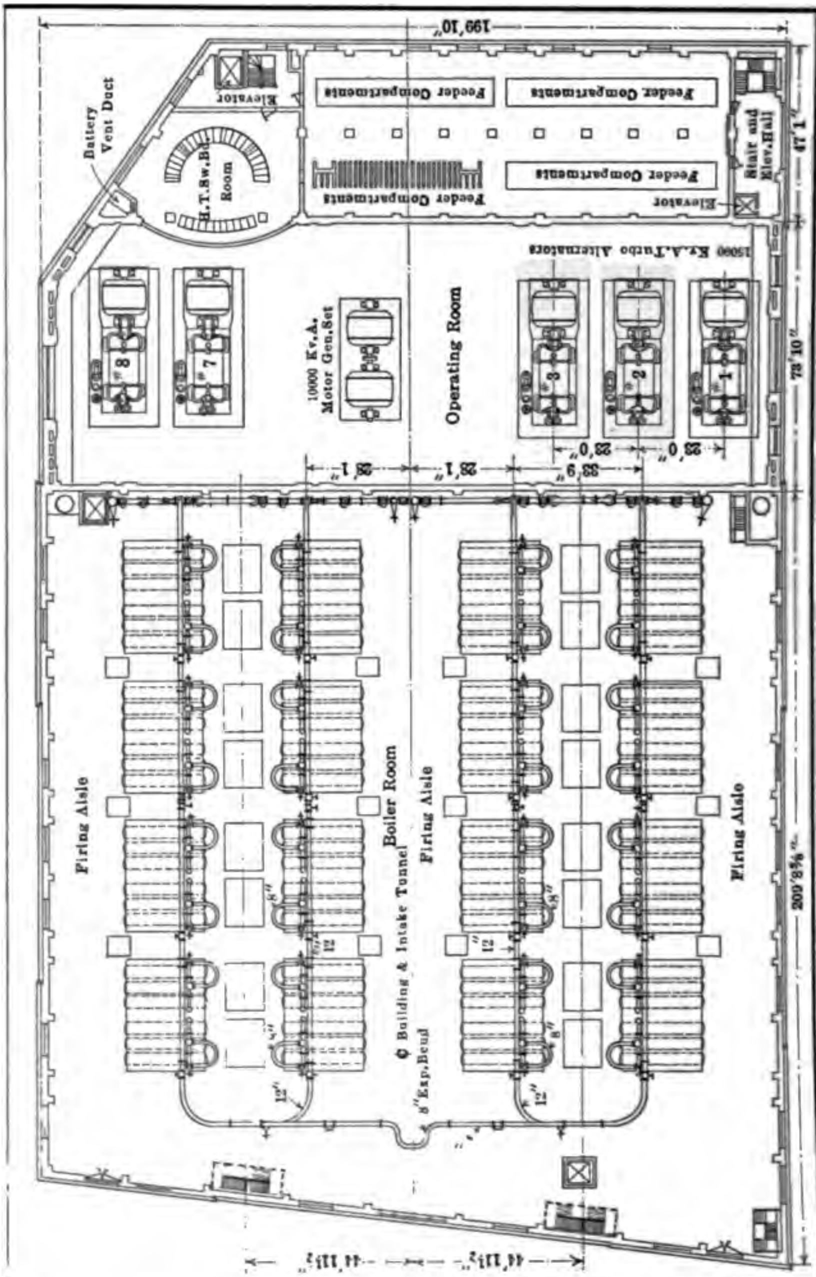


FIG. 158.—201st St. Power House, United Electric Light and Power Co., New York.

For known conditions the fuel consumption should be determined by using the B.t.u. value of the fuel and a proper boiler and furnace efficiency.

Some idea of the average coal consumption of plants of different sizes may be obtained from the following table, based on the coal per horsepower-hour.

COAL PER HORSEPOWER PER HOUR

Simple non-condensing:							
Engine horsepower	2	3	4	6	8	10	12
Total coal, pounds	13.0	10.5	8.5	7.9	7.6	7.4	7.25
Engine horsepower	14	15	20	30	40	50	75
Total coal, pounds	7.0	6.5	6.0	5.5	4.75	4.5	4.0
Simple condensing:							
Engine horsepower	10	12	14	15	20		
Coal, running times, pounds	6.1	5.9	5.7	5.25	4.80		
Total coal, pounds	7.0	6.75	6.50	6.00	5.50		
Engine horsepower	30	40	50	75	100		
Coal, running times, pounds	4.60	4.20	3.75	3.40	3.10		
Total coal, pounds	5.25	4.75	4.25	3.70	3.50		
Compound condensing:							
Engine horsepower	100	200	300	400	500	600	700
Coal, running times, pounds	2.75	2.45	2.40	2.35	2.30	2.25	2.20
Total coal, pounds	3.15	2.85	2.75	2.70	2.65	2.60	2.55
Engine horsepower	800	900	1,000	1,500	2,000		
Coal, running times, pounds	2.15	2.10	1.95	1.80	1.75		
Total coal, pounds	2.50	2.45	2.25	2.05	2.00		

Kent states[1] that small engines and engines with fluctuating loads are usually very wasteful of fuel. The following figures, illustrating their low economy, are given by Professor Unwin, *Cassier's Magazine*, 1894.

SMALL ENGINES IN WORKSHOPS IN BIRMINGHAM, ENGLAND

Probable i.hp. at full load	12	45	60	45	75	60	60
Average i.hp. during observation	2.96	7.37	8.2	8.6	23.64	19.08	20.08
Coal per i.hp. per hour during observation, pounds	36.0	21.25	22.61	18.13	11.68	9.35	8.50

It is largely to replace such engines as the above that power will be distributed from central stations.

Labor.—This charge should include the wages of all stokers, oilers, engineers, laboratory men, switchboard operators, electricians, clerks, janitors, watchmen and such portion of superintendence as is given to the station.

The wages of all men employed on repairs should be charged to maintenance.

[1] "Mechanical Engineers' Pocket-book," p. 964.

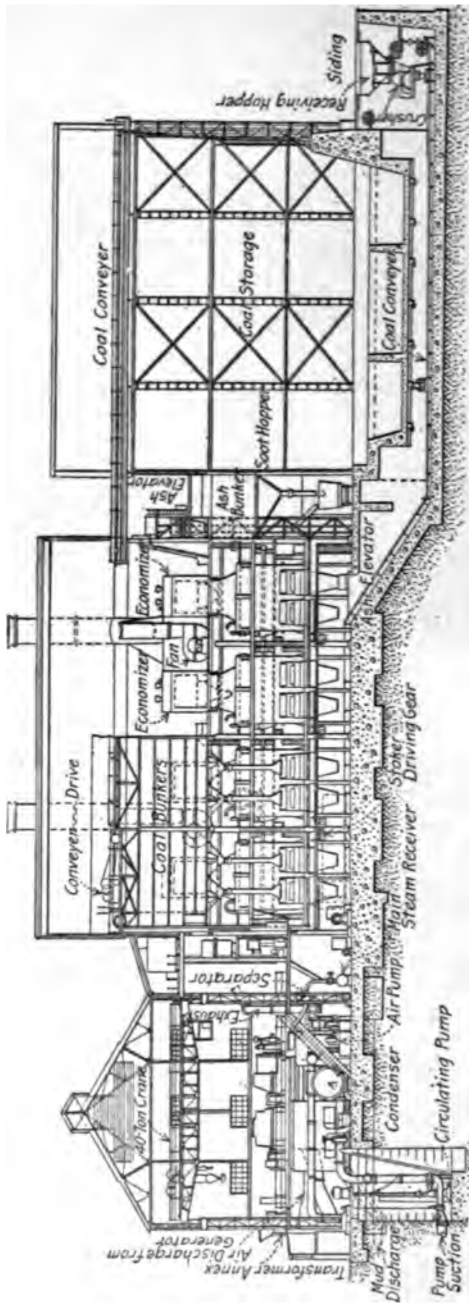


FIG. 159.—Longitudinal section Dunstan Power Station, Newcastle on Tyne.

The cost of attendance will depend upon the size of the plant, in general, being less for a large plant, *i.e.*, relatively less. The cost of engine attendance is greater for high-speed than for Corliss, and is also increased by the introduction of compounding or condensing engines.

The salaries paid engineers vary from $60 per month to $150 for ordinary plants.

Firemen receive from $50 to $90 per month, the average being about $65 for 12-hr. days.

Coal passers and ash wheelers receive about $30 to $55 per month.

In New York and Philadelphia firemen and coal passers receive in the neighborhood of $2 to $2.25 per 8-hr. day.

In general the yearly (3,080) hours cost of attendance will run about:

For simple non-condensing plants:								
Engine horsepower	2	3	4	6	8	10	12	14
Attendance	$99	$109	$116	$136	$154	$173	$184	$194
Engine horsepower	15	20	30	40	50	75		
Attendance	$202	$230	$287	$338	$390	$520		
For simple condensing plants:								
Engine horsepower	10	12	14	15	20	30		
Attendance	$178	$190	$202	$210	$238	$297		
Engine horsepower	40	50	75	100				
Attendance	$350	$405	$535	$670				

One man attends engine, fires boiler and is supposed to do other work besides. On the 10-hp. plant one-half of his time is charged to attendance and three-fourths of his time on the 100-hp. plant.

For compound condensing plants:						
Engine horsepower	100	200	300	400	500	600
Number of men and wages	1 at $16	1 at $16 1 at $7	1 at $16 1 at $7	1 at $16 1 at $10 1 at $7	1 at $16 1 at $13 1 at $7	1 at $17 1 at $13 1 at $10
Total attendance	$800	$1,220	$1,220	$1,760	$1,930	$2,100
Engine horsepower	700	800	900	1,000	1,500	2,000
Number of men and wages	1 at $17 2 at $22 1 at $10	1 at $18 2 at $22 1 at $10	1 at $18 2 at $25 1 at $10	1 at $19 2 at $26 2 at $20	1 at $22 3 at $36 2 at $20	1 at $25 4 at $50 2 at $20
Total attendance	$2,650	$2,700	$2,930	$3,480	$4,400	$5,200

These costs are below the average for the larger sized plants.

The following figures represent the cost of attendance for a large electric steam-turbine central station.

The wages involved in the cost of power delivered by one unit (14,000-

kw. Curtis turbine, 8 B. & W., boilers of 5,000 sq. ft. of heating surface each and auxiliaries) to the switchboard are:

General Engineering Force:

	Cost per day
One chief engineer at $250 per month, 1/6 of his time	$1.40
One assistant chief engineer at $200 per month, 1/6 of his time	1.11
One chief electrician at $200 per month 1/6 of his time	1.11
One assistant chief electrician at $150 per month, 1/6 of his time	0.83
Three load despatchers at $100 per month, 1/6 of their time	1.66
One boiler-room foreman at $100 per month, 1/6 of his time	0.55

Operators for One Generating Unit:[1]

Three watch engineers at $4 per day of 8 hr	$12.00
Three oilers at $2.50 per day of 8 hr	7.50
Three switchboard attendants at $2.50 per day of 8 hr	7.50
Three firemen at $2.50 per day of 8 hr	7.50
Three water tenders at $2.50 per day	7.50
One boiler washer at $2.50 per day of 8 hr	2.50
One pipe fitter at $3 per day of 8 hr	3.00
One pipe fitter helper at $1.50 per day of 8 hr	1.50
Four laborers at $2 for coal handling	8.00
Total	$63.66

Herrick has given a table of interest in connection with the labor item.

Plant	Rating, kw.	Output, kw.-hr.	Total station wages	Labor per kw.-hr., cents	Total station cost, kw.-hr., cents	Number of station employees
A	6,000	8,776,165	$25,937	0.296	1.21	22
B	5,000	6,043,204	20,920	0.346	1.23	20
C	4,000	5,400,192	19,429	0.360	1.24	28
D	2,000	3,288,623	9,954	0.302	1.42	11
E	2,000	4,305,003	9,663	0.224	1.27	13
F	1,250	1,470,066	6,844	0.465	1.56	8
G	950	1,479,898	8,771	0.595	2.05	7
H	700	889,760	6,669	0.750	2.34	8
I	630	730,458	5,017	0.685	1.80	6

Plant	Kw. per station employee	Wages per kw. station capacity
A	272.0	$4.31
B	250.0	4.18
C	136.0	5.10
D	182.0	4.97
E	154.0	4.83
F	157.0	5.45
G	136.0	9.25
H	87.5	9.52
I	105.0	7.95

[1] In the plant upon which these figures are based, there is a watch engineer and oiler employed to look after each unit.

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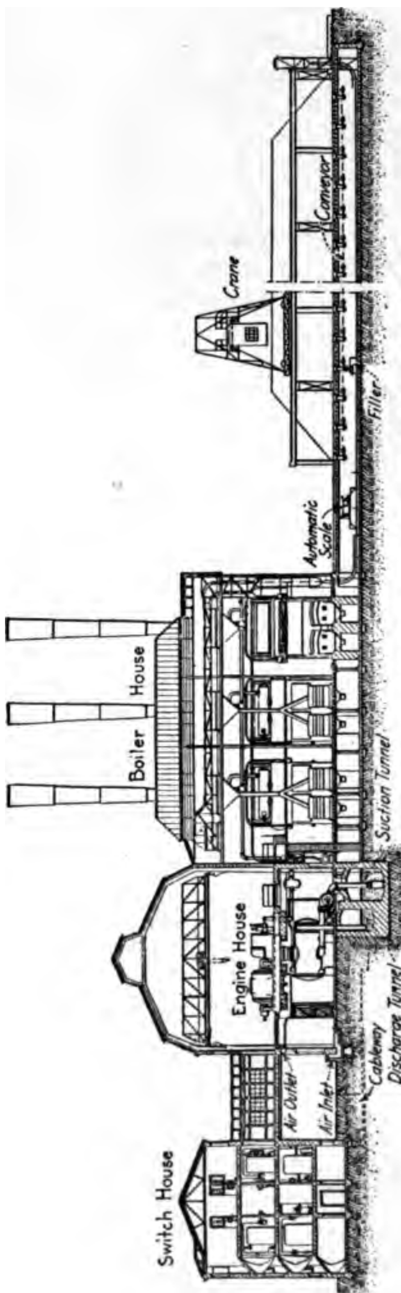


Fig. 160.—Longitudinal section Eberswalde Station Markische Elektricitats Werke.

custom of some oil companies to contract to furnish all the oil needed at a certain price per kilowatt-hour generated. The prices have varied between 0.01 and 0.02 ct. per kilowatt-hour.

White waste of good grade may be purchased in 150-lb. bales at about 9 cts. per pound (variation 7 to 11 cts.). This item may amount to considerable unless care is taken.

Relatively large financial savings may be made by using a waste and oil separator. By this means waste may be used six or eight times and large amounts of oil are recovered which, after filtering, may be reused.

One small station of about 750- or 1,000-hp. capacity installed such a separator at a cost of $150, and saved $90 in oil and waste the first month. The average saving per month in this plant is over $100.

In the larger stations washable cheesecloth towels have replaced waste.

Supplies include such small articles as packings, small pipes, valves and fittings, tools, wrenches, gaskets and other small articles which must be kept in stock. They should include laboratory supplies, stationery, janitors' supplies, and other items of this kind.

These three items are usually lumped together and form a small part of the station cost in a large plant.

One consulting engineer reports the yearly (3,080 hr.) cost of oil, waste and supplies to run approximately as follows:

Simple non-condensing:								
Engine horsepower	2	3	4	6	8	10	12	14
Oil, etc.	$13.20	$14.30	$14.30	$17.60	$20.00	$22.00	$23.80	$25.80
Engine horsepower	15	20	30	40	50	75		
Oil, etc.	$26.50	$31.20	$41.50	$51.00	$61.50	$85.50		
Simple condensing:								
Engine horsepower	10	12	14	15	20	30		
Oil, etc.	$22.80	$24.80	$26.70	$27.60	$32.50	$43.00		
Engine horsepower	40	50	75	100				
Oil, etc.	$53.00	$64.00	$89.00	$114.00				
Compound condensing:								
Engine horsepower	100	200	300	400	500	600	700	
Oil, etc.	$143.00	$205.00	$240.00	$285.00	$315.00	$350.00	$385.00	
Engine horsepower	800	900	1,000	1,500	2,000			
Oil, etc.	$420.00	$445.00	$470.00	$600.00	$685.00			

These costs are below the average for the larger size plants.

Water.—This expense will be relatively small with a condensing station even where city water is used. In many cities the cost of water for manufacturing purposes is 40 cts. per 1,000 cu. ft. or $5.35 per 1,000 gal. For New York City the price is $1 per 1,000 cu. ft. Where fresh water is used for condensation the feed water can usually be taken from the tail pipe and will cost nothing. With turbine stations and surface condensers the make-up is very small and the water item is smaller than the oil item. Caution should be exercised in using Artesian or other well water as feed water if its chemical composition is not known.

PROBLEMS

56. Estimate the cost of a small power plant of 125 i.hp. The estimate is to be based on the assumption that the engine and boiler are not more than 20 ft. apart; that water supply is brought into the engine room by customer; that exhaust valve and safety-valve exhaust are to be carried outside of the engine room a distance of not more than 20 ft.; that sewer connection is made in engine room to which drip pipes and blowoff pipes may be carried.

1. 125-hp. simple non-condensing engine, on cars.
2. Freight to destination, estimated.
3. Cartage and handling to position, about $5 per ton.
4. Foundation.
5. Boiler, horizontal fire-tube horsepower.
6. Freight.
7. Handling.
8. Setting.
9. Iron stack.
10. Erecting.
11. Feed pump.
12. Feed-water heater.
13. Pipe connections, steam, exhaust and water pipe.
14. Pipe covering for all steam pipe.
15. Man to superintend erection (customer furnishing all laboring help in handling heavy pieces) 5 days at $5 per day.
16. Railroad expenses from factory and board.
17. Add to above for contingencies, 5 to 10 per cent.
18. Add consulting engineer's or agent's commission, if any.
19. Add expense of test after erection, if required.

57. An equipment is to be selected for a steam power plant capable of developing 750 hp. Three estimates are to be made as follows:

A. Two 300-hp. compound, high-speed, non-condensing engines and one 150-hp. compound, high-speed, non-condensing engine. Return tubular boilers. Equal units. Two in operation, and one in reserve. Flat grates, hand-fired. Equivalent evaporation 7.5 lb. water per pound of coal.

B. Two 300-hp. compound, condensing Corliss engines and one 150-hp. compound, condensing high-speed engine. Water-tube boilers and mechanical stokers. Same arrangement of boilers as in "A." Equivalent evaporation 8 lb. water per pound coal.

C. Two 375-hp. compound, condensing Corliss engines. Boiler equipment and conditions as in "B."

Determine the best plant to install:

(a) If water is purchased and wasted to sewer.
(b) If water costs nothing but pumpage.

NOTE.—Standby losses need not be considered in this problem.

Investment "A."

1.	Two 300-hp. engines, erected	$
2.	Foundations for two engines	
3.	One 150-hp. engine, erected	
4.	Foundation for engine	
5.	—— boilers, —— hp. each, including setting	
6.	Brick stack	
7.	Flues	
8.	Two feed pumps	
9.	Two feed-water heaters	
10.	Piping	
11.	Boiler house	
12.	Engine house	
13.	Coal pocket	
14.	Incidentals	
	Total	$

Investment "B."

1.	Two 300-hp. engines and condensers, erected	
2.	Foundations for two units	
3.	One 150-hp. engine and condenser, erected	
4.	Foundation	
5.	—— boilers, —— hp. each, including setting	
6.	Brick stack	
7.	Flues	
8.	—— mechanical stokers	
9.	Two feed pumps	
10.	Two feed-water heaters	
11.	Piping	
12.	Boiler house	
13.	Engine house	
14.	Coal pocket	
15.	Incidentals	
	Total	$

Investment "C."

1.	Two 375-hp. engines and condensers, erected	
2.	Foundations for two units	
3.	—— boilers, —— hp. each, including setting	
4.	Brick stack	
5.	Flues	
6.	—— mechanical stokers	
7.	Two feed pumps	
8.	Two feed-water heaters	
9.	Piping	
10.	Boiler house	
11.	Engine house	
12.	Coal pocket	
13.	Incidentals	
	Total	$

Estimated Operating Cost.—Service 10 hr. per day, 308 days per year. Load on engine = 85 per cent. of full-load rating.

Plant "A"

	Amount per day	$ per day
1. Coal: Tons for two 300-hp. engines		
Tons for one 150-hp. engine		
Tons for auxiliaries and leakage		
2. Attendance: —— engineers at $——		
—— firemen at $——		
—— coal passers at $——		
3. Oil: Cylinder oil for all engines		
Engine oil for all engines		
4. Waste and supplies		
5. Water: M cu. ft. for two 300-hp. engines		
M cu. ft. for one 150-hp. engine		
M cu. ft. for auxiliaries and leakage		
6. Maintenance, 6 per cent. on $——		
Operating expenses only	$——	——
7. Fixed charges at —— per cent. on $——		
Total operating cost and fixed charges	$——	——

Plant "B"

	Amount per day	$ per day
1. Coal: Tons for two 300-hp. engines		
Tons for one 150-hp. engine		
Tons for auxiliaries and leakage		
2. Attendance: —— engineers at $——		
—— firemen at $——		
—— coal passers at $——		
3. Oil: Cylinder oil		
Engine oil		
4. Waste and supplies		
5. Water: M cu. ft. for auxiliaries and leakage		
M cu. ft. for circulating water		
6. Maintenance, 6 per cent. on $		
	——	——
Operating expenses only $		
7. Fixed charges at —— per cent. on $——		
	——	——
Total operating cost and fixed charges $		

Plant "C"

	Amount per day	$ per day
1. Coal: Tons for two 375-hp. engines		
Tons for auxiliaries and leakage		
2. Attendance: —— engineer at $——		
—— firemen at $——		
—— coal passers at $——		
3. Oil: Cylinder oil		
Engine oil		

4. Waste and supplies....................................
5. Water: M cu. ft. for auxiliaries and leakage..............
M cu. ft. for circulating water.......................
6. Maintenance, 6 per cent. on $——......................

Operating expenses only...............................$
7. Fixed charges at —— per cent. on $——................

Total operating cost and fixed charges..................

Summary

	A	B	C
Investment..			
Excess cost over "A"............................			
Total operating cost per year......................			
Saving in operation cost per year over "A"..........			
Time to make up difference in first cost by saving in operating expenses..............................			
Total indicated horsepower-hours..................			
Cost per indicated horsepower-hour.................			

58. Assuming the same types for the installation as in problem 56 estimate the operating cost for 24 hr. service for 365 days per year.

CHAPTER XIII

S l. . l

VARIABLE LOAD ECONOMY

The making of power, up to a very few years ago, was entirely a local business, each mill, shop or factory was located at the power supply, or when steam was used the engine was located in or near the factory building. The transmission to the machines was by shafting, gears or belts. About 60 years ago rope transmission and a little later hydraulic and compressed-air transmissions were introduced in certain localities in order that the power might be used at some distance from the point of generation. About 30 years ago electrical transmission came into use

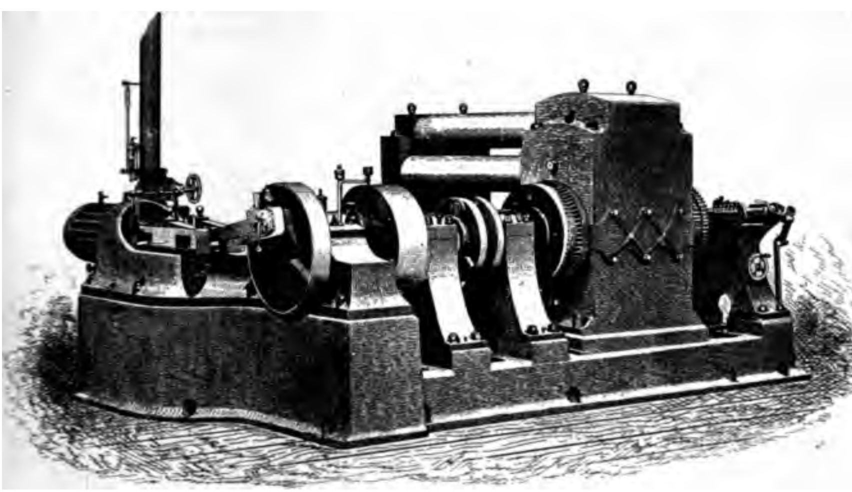


FIG. 161.—Jumbo dynamo and Armington & Sims engine. The first Central Station unit, built June, 1881.

and with it the central station became an established fact. Both the hydraulic transmission, so well applied at London and Geneva, and the compressed-air transmission, as it was used in Paris, might have been capable of development, but the essential convenience and economy of the electrical transmission and drive soon gave it first place, and today when one speaks of the central-station supply of power the electrical system is implied.

The electrical transmission of power has its disadvantages, the chief being the impossibility of storing power in any quantity. Electricity must be used as generated. This means that there will be a varying load on the central station at all times and that the generators must be large enough to generate the maximum power required at one time and also must be run at something below the best efficiency most of the time. This variation of load is best shown by a load curve, plotted with time

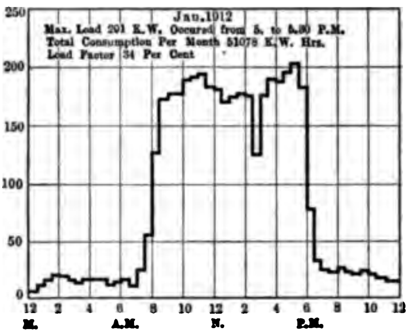


FIG. 162.—Load curve of average size department store.

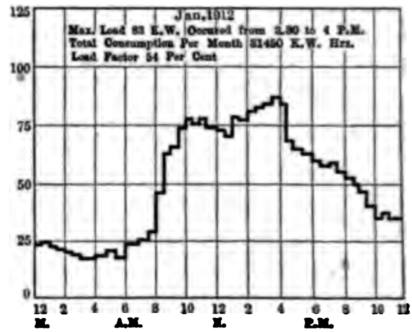


FIG. 163.—Load curve of U. S. Gov't. Post Office Building.

for the abscissæ and power for the ordinates. Such load curves for central stations with various kinds of load are shown in the accompanying figures.

These load curves will vary from day to day and season to season, but for each kind of business a characteristic curve can be drawn.

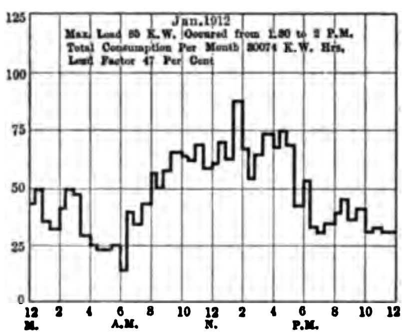


FIG. 164.—Load curve of newspaper plant.

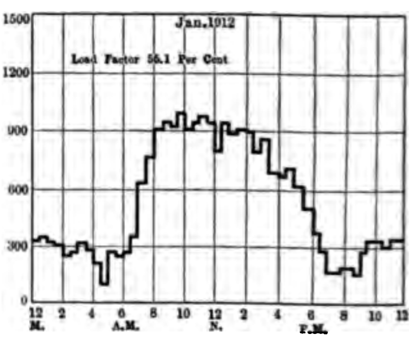


FIG. 165.—Combined load curve of nine large power consumers.

Through the courtesy of The Detroit Edison Co., Figs. 166 and 167 are presented showing the change in load distribution due to the recent change from central to eastern time in that city.

One blue print is plotted on clock time as a basis; the other is plotted on sun time. In each case the ordinates are based on the peak of the curve before the change of time, this peak being called 100 per cent.

In comparing these curves it should be noted that the change occurred in the late spring and that there would naturally have been an increase in the depth of the valley preceding the peak, as well as a decrease in the height of the peak itself.

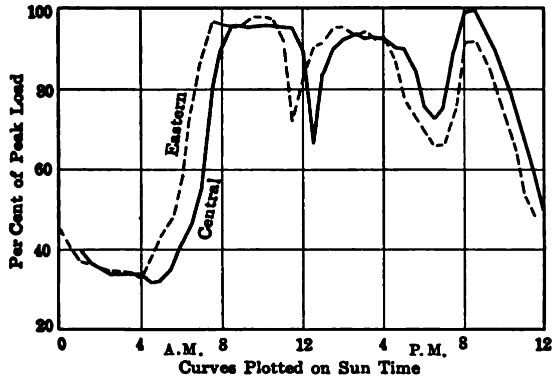


FIG. 166.—Detroit Edison Co. load curve showing effect of change of time.

On the basis of clock time the curves show that people apparently lived by the clock in the morning, but that they lived more by the sun in the afternoon and early evening, since the peak was delayed about 1 hr., the exact amount by which the clocks had been moved ahead.

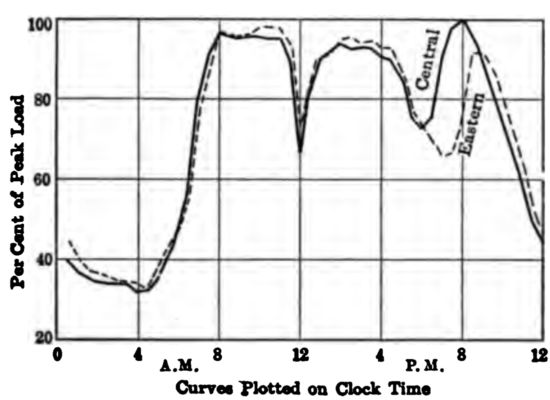


FIG. 167.—Detroit Edison Co. load curve showing effect of change of time.

This is shown in just the reverse fashion on the sun-time basis. Morning, noon and afternoon events are seen to be made earlier by about 1 hr. with respect to the sun, indicating living on the basis of clock time. On the other hand, the peak occurs at sun time, indicating practical independence of the clock for the evening events.

Load Factor.—For convenience in determining operating conditions, the average load per day, per month or per year, is calculated, and by comparing this average load with the maximum load in the same time period, a very valuable indication of the character of the load is obtained. The American Institute of Electrical Engineers in its standardization

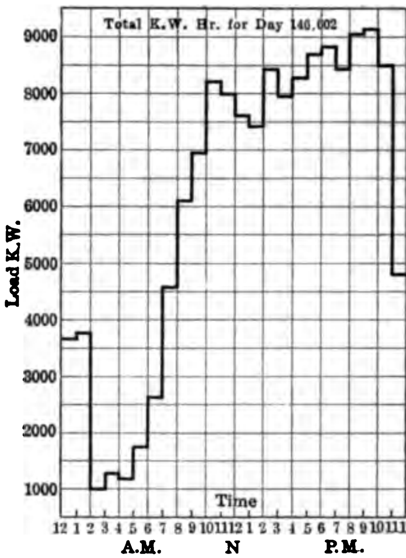


FIG. 168.—Load curve, July 4, 1910, Aurora, Elgin & Chicago R. R.

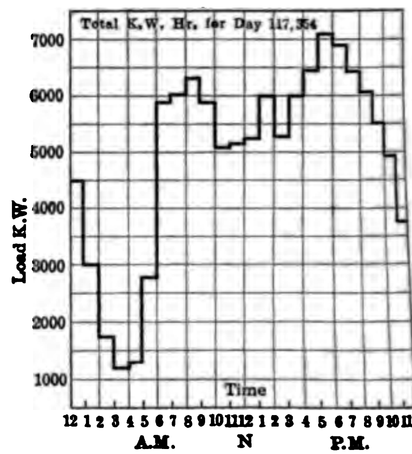


FIG. 169.—Typical weekday load curve, Aurora, Elgin & Chicago R. R.

rules defines "Load Factor" as the ratio of the average load to the maximum load during a certain period of time. The average load is taken over a certain interval of time, such as a day or a year, and the maximum is

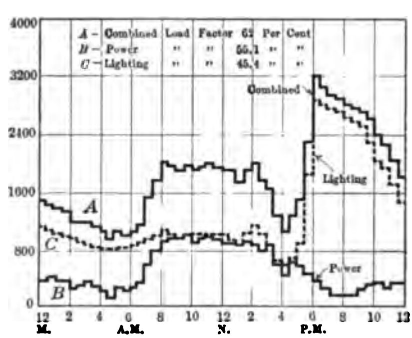


FIG. 170.—Winter load curve power and lighting.

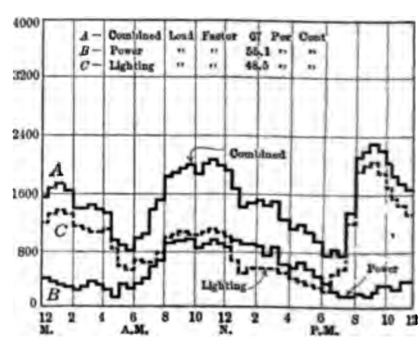


FIG. 171.—Summer load curve power and lighting.

taken over a short period, such as 15 min., or an hour, within that interval. In each case the interval of maximum load should be definitely specified. It is dependent upon the local conditions and the purpose for which the load factor is to be determined. För electric-light plants and for operating statistics it is usual to consider the load factor as the ratio of the average load for the day to the maximum 5-min. peak. In a street-railway plant where the momentary swings are larger, the period for determining the maximum may to advantage be made as large as 30 min. or evèn 1 hr., and the maximum peak taken as the average during that period. The yearly load factor is the average load for the year divided by the maximum during the year.

Station economies depend on load factors to a considerable extent and high load factors are uncommon, except in metallurgical plants, where a load factor of 90 per cent. may be at times attained. Railway plants give load factors varying between 3 per cent. and 50 per cent. Lighting stations rarely exceed 30 per cent. and the smaller lighting plants are sometimes as low as 3 per cent. Williams and Tweedy in "Commercial Engineering for Central Stations," give a list of 24 central stations with varying output from 900,000 kw.-hr. per year to over 32,000,000 kw.-hr. per year, in which the lowest load factor is 19.2, the highest load factor 36.5, and the average for the 24 plants 28 per cent. A station with a high load factor should have few units and large ones and the most economical apparatus will quickly pay for itself. A low load factor will mean smaller units and a larger number of them and the economy of at least half of the apparatus is of no great consequence, since it is only used a few hours every year. In some of the larger lighting plants as much as 97 per cent. of the output is generated on less than 50 per cent. of the apparatus, the remaining half of the machines are in use less than 60 hr. out of the 8,760 hr. of the year.

A. F. Strouse in the *Electric Journal* has given the following table of industrial load factors which may be useful:

	Per cent.		Per cent.
Boiler shops	10–20	Foundries	5–15
Shoe factories	15–25	Knitting mills	25
Breweries	45	Machine shops	5–25
Cement plants	60–90	Clay products	15–20
Coal mines	15–30	Tanneries	10–20
Cotton mills	20–30	Textiles (general)	25
Flour mills	20–25	Woodworking ships	5–30

Diversity Factor.—It will be noted that load factor is a measure of the load on the central-station system, and is independent of the type or kind of power generation. There is another factor which also deals with the system load, which is of great importance to the operator and designer. It has been noticed that while one piece of apparatus on an electrical supply main may occasionally take its maximum power, two such machines will not take double the power, because their maximums do not come at the same time. If we take the sum of the maximums of all the connected loads on the system and divide them by the maximum load on the system we have a factor which has been called the diversity factor, and in every case is greater than one, and in some systems may be as large as four or five. The National Electric Light Association, in 1912, changed the definition of this factor while retaining the name, as follows: "diversity factor is the ratio between the simultaneous demand of a number of individual services for a specified period, and the sum of the individual demands of those services for the same period." This definition expressed as a fraction or as a percentage (never greater than one) is now universally accepted. The diversity factor of a purely lighting load may be as low as 25 per cent. With motor loads the factor is 50 per cent. or higher.

Gear (*Electrical World*, Nov. 10, 1910) gives the following table of block diversity factors and load factors for three classes of electrical service:

ANALYSIS OF CUSTOMER'S DIVERSITY FACTORS

Group	Number of customers	Kw. connected per customer	Sum of consumer's maxima	Maximum of group	Diversity factor	Average consumers load factor	Group load factor
	Residence Lighting						
Block A.....	34	0.53	12	3.6	0.3	7.0	23.3
Block B.....	185	0.53	68	20.0	0.294	7.0	23.8
Block C.....	167	0.87	93	28.0	0.302	7.3	24.0
Average.....	128	0.68	57	17.2	0.299	7.1	23.9
	Commercial Lighting						
Group D....	46	1.28	46	33.0	0.714	13.0	18.0
Group E....	79	0.74	36	26.0	0.714	11.0	16.0
Group F....	160	0.53	62	41.0	0.662	10.0	15.0
Group G[1]...	221	2.70	403	270.0	0.675	13.0	19.0
Average.....	95	0.70	48	33.0	0.685	10.8	15.7
	General Motor Service						
Group H....	29	0.1 hp.	30 kv.a.	21 kv.a.	0.7	15.0	21.0
Group I.....	18	3.3	40	25	0.625	16.0	26.0
Group J.....	11	11.8	90	65	0.719	18.0	28.0
Group K.....	25	6.0	100	70	0.7	21.0	30.0
Average.....	21	4.5	65	45	0.695	15.5	26.0

A and B, apartments; C, apartments and residences; D, small stores, saloons, restaurants; E, small stores above; F, apartments above stores, lodge halls, etc.; G, office building; H, I, J, K, mostly clothing manufacturers.

[1] G is not included in average of group.

Use Factor.—The designing engineer uses a factor analogous to load factor which is known as the use factor and sometimes as the utility factor. This may be defined as the ratio between the average power sent out by the station to the maximum 24-hr. rating of the station. This factor is in all cases lower than the load factor by the amount of reserve. The load factor and diversity factor tell what part of the installation is used and how much it is used, but they do not show the capacity of the generating station. The use factor does this and may be used for design.

Readiness to Serve.—"The readiness to serve" item plays a large part in station economy. If we know what the load will be 15 min. or ½ hr. ahead due preparations can be made with ease and certainty. Where this is not known machinery must be run light in anticipation of a load that may never come. That such loads do come at times is shown by the "thunderstorm peaks," which may increase the load 50 per cent. in 5 min.

Central Station Design.—The power-station problem, as presented to the designing engineer, usually takes the form of a deductive analysis,

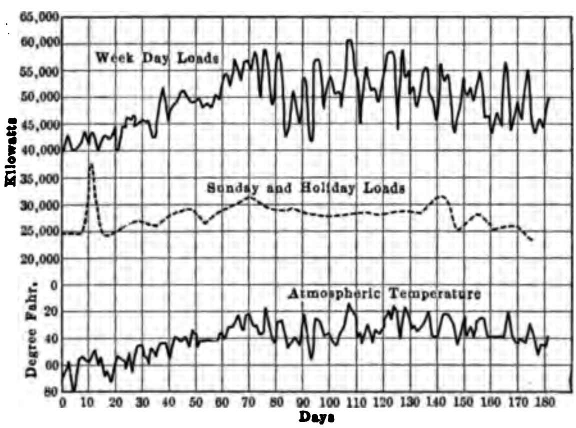


FIG. 172.—Boston Elevated Ry. load curve showing effect of temperature on a railway load.

starting from several given conditions which are at once the basis and the limitation of the problem. These conditions may be scheduled as follows: First, method of power generation; second, the type of load, that is, the load factor and diversity factor; third, the locality over which the power is to be distributed and used; fourth, various restrictions depending on peculiar physical, political or commercial considerations. Of these conditions the last two are always beyond the control of the designing engineer, while the first two are more or less directly under his control, and should be the subject of careful studies to insure the best solution. Under location, it is necessary to consider the territory to be covered, the distribution of population, lines and distribution of travel and manufacturing and commercial localities. If the central station is to supply a lighting load, a power load, a railway load or any combination of these, the locality must be studied with reference to these conditions. The various political, commercial and artistic restrictions, under the fourth category, must be taken into consideration in the analysis, leading up to the centers of distribution of population. With these studies it will be possible to approximate quite closely to what may be expected regarding the use of current at all times in the 24 hr., and a load curve may be laid out and the load factor ascertained. The proper location of the central station may then be picked out with reference to ease and economy of distribution. During the foregoing analysis the following points will have been considered and tentatively settled; the maximum load, the average yearly and daily load, the system of distribution, the location of the central station, the location of the various distribution lines, substations, feeders and mains, and finally the types of prime movers. The engineer is now ready to roughly design the various parts of the system and to make the preliminary layouts and estimates from which the investor must decide as to whether the project will be commercially successful.

Standby Losses.—In the commercial operation of a plant under changeable load conditions it is necessary to retain steam pressure on boilers

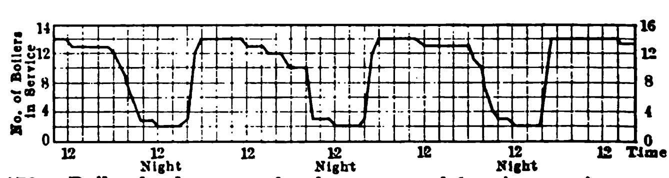


FIG. 173.—Boiler load curve of a large central heating station. (Josse.)

which are otherwise idle in order to provide sufficient reserve capacity for sudden power demands and to keep up steam pressure during periods when the plant is shut down. Owing to radiation, leakage, and other losses, this requires considerable fuel.

Take the case of an office-building plant operating 18 hr. per day. There are 6 hr. during which no power is being turned out. It is therefore necessary either to keep up sufficient fire to maintain the normal boiler pressure during this period or else to close all dampers and openings and make up the reduced pressure before starting up the plant. Again, during the busy hours of the day—say from 5 to 6 o'clock when all elevators are running and, if in the winter time, nearly all the lights are burning—the power requirement reaches its maximum and all the boilers are operating at a high capacity. But during the light run—say 8 or 9 o'clock on a bright summer morning—only part of the boilers are required, and (if the plant be a large one) probably only one of the generating sets. And, as in the case when the plant is completely shut down, steam pressure must be either maintained or reëstablished before the boilers are brought into service. Then there are the losses due to turning over reserve engines preparatory to their being "cut in," and the warming up of idle units, etc. It will be noted that all of this requires fuel while not adding in any degree to the power output. The fuel losses caused by these conditions are known as "standby losses" and are common to every plant.

Their magnitude depends upon the load factor, hours of operation per day, number of units, design and construction of plant, kind and effectiveness of non-conducting covering, climatic conditions, etc., and is, therefore, a very difficult factor to accurately determine.

For first-class plants the standby losses are probably about 6 per cent. of the fuel required at normal boiler rating over a period of time

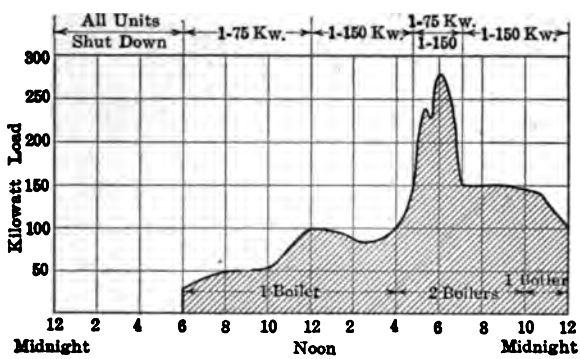


FIG. 174.—Load curve of office building.

equal to the standby period. The average is probably nearer 10 per cent.

The simplest manner in which to explain this standby loss is by reference to an actual example (see Fig. 174).

The plant consists of two units of 75 kw. each (one of which is used as a reserve unit only) and one of 150 kw. rated capacity; and three water-tube boilers of 150 hp. each (one being for reserve purposes). The generating sets are high-grade, tandem-compound, self-oiling, automatic, direct-connected, non-condensing, piston-valve units.

The general method of determining probable standby losses is first to establish the points on the load curve where different boilers should be cut in and out, thus dividing the day into several periods during each of which certain boilers are operated. From this is obtained the total boiler capacity idle and the number of hours of such idleness from which the standby loss may be approximated in accordance with the above.

This, of course, involves a knowledge of boiler efficiencies and engine economies.

For the case in hand the exhaust steam from the engines is more than sufficient for furnishing heat for the building so that all the steam generated in boilers is used by the engine and auxiliaries. Now, in order to determine the proper time for cutting in or out various boilers it is first necessary to predetermine just when different generating units will be required in service. It will be noted that in the example under considera-

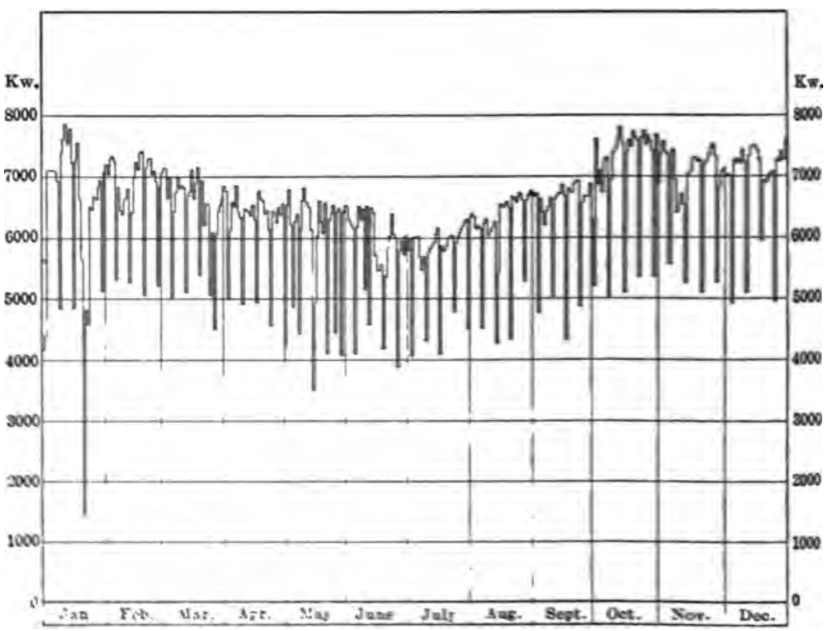


FIG. 175.—Curve of daily maximum demands. Wangen, a. A.

tion the entire day of 24 hr. has been divided primarily into five divisions, as follows:

1. From 12:00 midnight to 6:00 a.m.
2. From 6:00 a.m. to 12:00 noon.
3. From 12:00 noon to 4:45 p.m.
4. From 4:45 p.m. to 7:00 p.m.
5. From 7:00 p.m. to 12:00 midnight.

From midnight until 6 o'clock in the morning the entire plant is out of service.

The second period—that during the morning hours—is well taken care of by one 75-kw. set.

During the third of these periods the maximum output is 150 kw.

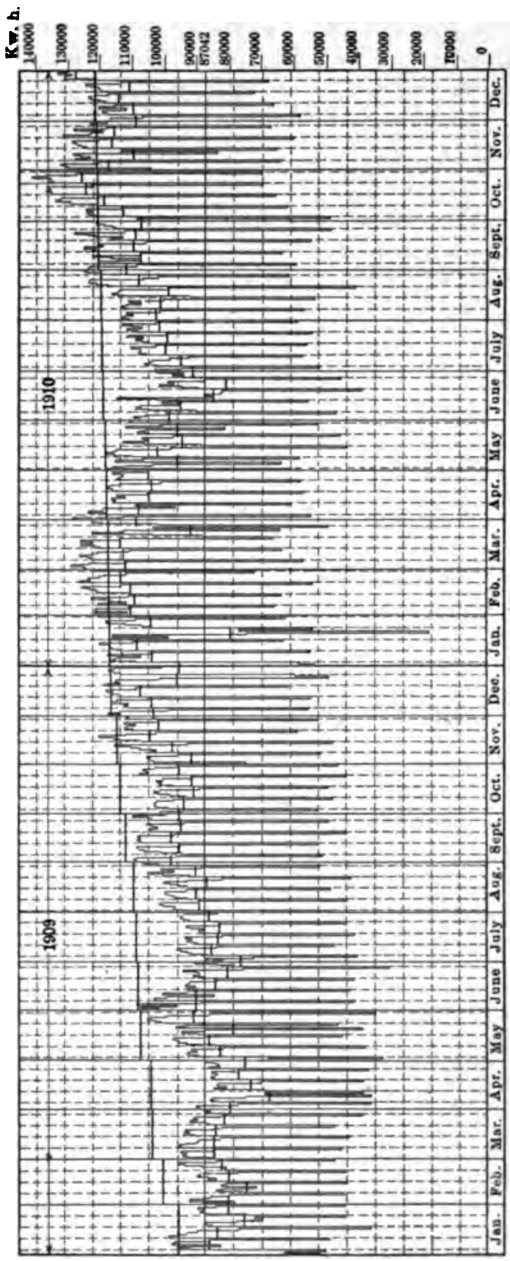


FIG. 176.—Curves of daily output. Wangen a. A.

and the average about 100 kw. One unit is kept in service, viz., the 150-kw. machine.

During the fourth period the load rapidly increases to a maximum of 275 kw., the average being about 180 kw. One 150-kw. and one 75-kw. unit take care of these conditions operating at a slight overload for about 1 hr.

The fifth period is taken care of quite economically by the original 150-kw. machine, the smaller unit having been shut down at 7:00 p.m.

The determination of the amount of steam required and therefore the number of boilers necessary at any moment is now an easy matter. Say for instance, the plant is started up at 6:00 a.m. with one boiler in service. The steam requirements are easily within the capacity of this boiler until the load suddenly increases late in the afternoon. At 4:00 p.m. 100 kw. is being generated, the unit being operated at about two-thirds of its rated load at that instant. Assuming the steam consumption to be 25 lb. per indicated horsepower-hour and a combined engine and generator efficiency of above 90 per cent., there will at that moment be required steam at the rate of approximately 4,300 lb. per hour. This allows about 15 per cent. for auxiliaries, etc., and is roughly equivalent to 145 boiler hp. This is a very small load for a good water-tube boiler of 150-hp. capacity, but the load increases very suddenly after 4:00 p.m. so that it is necessary to cut in the second boiler at that time.

At 6 o'clock the load has reached its maximum of 275 kw. and there is then required for the entire plant steam at the rate of about 12,000 lb. per hour or 400 boiler hp. This condition is, however, only instantaneous, the demand increasing and decreasing very rapidly before and after this time. Two 150-hp. boilers are, therefore, easily capable of caring for these conditions until about 10:00 p.m. when one boiler is cut out, leaving the remaining one in operation until the plant is shut down at midnight.

It is easily seen that the day is further divided into four periods of operation in the boiler room which may be summarized as follows:

1. From 12:00 midnight to 6:00 a.m., no boilers used.
2. From 6:00 a.m. to 4:00 p.m., one boiler used.
3. From 4:00 p.m. to 10:00 p.m., two boilers used.
4. From 10:00 p.m. to 12:00 midnight, one boiler used.

From this is developed the number of boiler horsepower idle during the day, neglecting the reserve boiler which is always idle and which is not considered for the reason that it is not kept under steam and therefore uses no fuel.

There are two boilers idle from 12:00 midnight to 6:00 a.m.—or 6 hr. each.

There is one boiler idle, but under steam, from 6:00 a.m. (when plant is started up) until 4:00 p.m.—or 10 hr.; there is one boiler idle from 10:00 p.m. to 12:00 midnight or 2 hr.

This is equivalent to a 150-hp. boiler idle (2 × 6) + 10 + 2 = 24 hr.

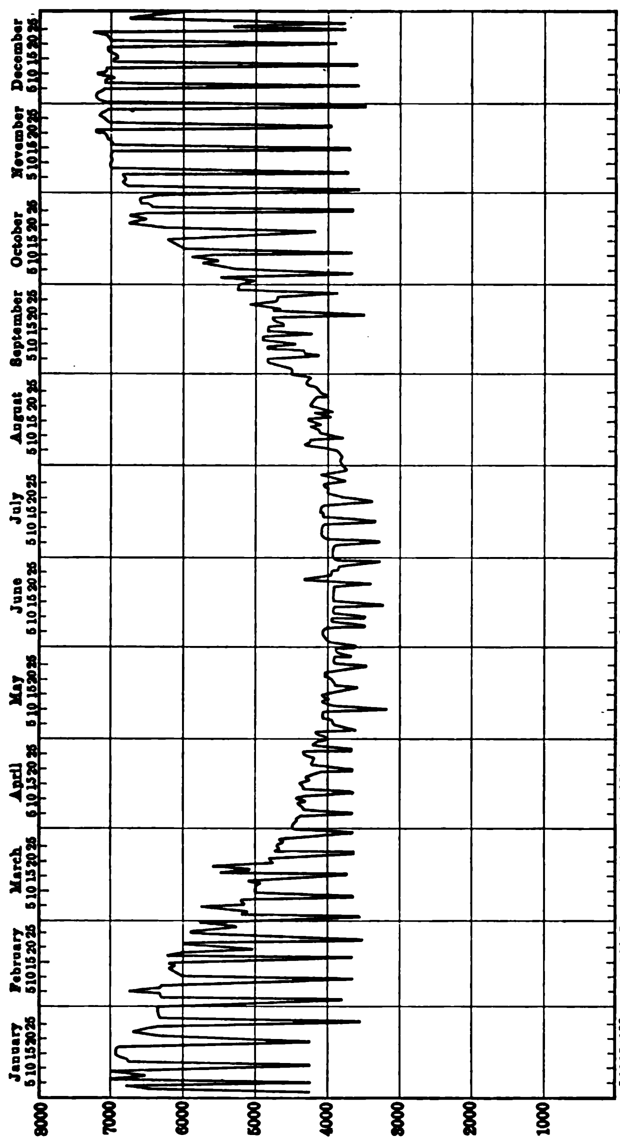


FIG. 177.—Curve of maximum loads (light and power only) city of Geneva, Switzerland, Chevres Power Station.

If this one 150-hp. boiler were actually operating at full load for 24 hr. there would be developed 150 × 24 or 3,600 boiler hp.-hr.

If oil were used as fuel and if its calorific value be 18,500 B.t.u. per pound and the boiler efficiency 70 per cent., there would be required

$$(3{,}600 \times 34.5 \times 970) + (336 \times 18{,}500 \times 70)$$

barrels of oil, or 27.5 bbl. on the basis of 336 lb. per barrel. The standby loss is about 6 per cent. of this or about 1.65 bbl. per day.

As a further illustration of standby losses, consider the following abstract of a letter[1] from John Hunter, Chief Engineer of Power Plants, Union Electric Light and Power Co., St. Louis, describing the practice at the Ashley Street Station in banking fires.

"There are two general conditions under which boilers are maintained in a banked condition. First, boilers only required for 2 hr. at the peak in the afternoon and known as a short bank, where the minimum amount of coal is burned and where the steam pressure in the boiler is allowed to drop as low as 60 lb. To maintain a banked fire on a 100-sq. ft. chain grate in this condition requires 130 lb. of coal per hour. The other condition is what is known as a long bank, in which about 450 lb. of coal per hour are burned on the grate and the boiler is kept on the header, but delivers only a small amount of steam. These boilers are used for varying load.

Under present condition of operation at Ashley Street, three boilers more than the number required for carrying the steam load at any time are always carried on a long bank. With 22 boilers under fire the maximum number required on the load during the peak is 19 boilers. On the 528 boilers hours each day there are 223, or 42.2 per cent., which are banking hours.

Seventy-two of these banking hours are at the rate of 450 lb. of coal per hour (boilers on the long bank), consuming a total of 32,400 lb. of coal. The remaining 151 banking hours (boilers on the short bank), are at the rate of 130 lb. per hour, consuming a total of 19,630 lb.

The total amount of coal used for banking during the 24 hr. is 52,030 lb. or 26 tons, which at a cost of $1.10 per ton amounts to $28.60 per day. The amount of coal for banking amounts to 5.1 per cent. of the total consumption for the station.

When notice is given of a storm existing anywhere along the transmission line, or when there is any possibility of an interruption in the service, standby boilers are started up and kept in readiness to pick up the load. The banking consumption on the standby boilers at such times will run about 450 lb. of coal per hour."

Carrying Peak Loads Economically.—The sources of power for peak loads are:

(*a*) Storage batteries.
(*b*) Purchased power.
(*c*) Hydroelectric power.
(*d*) Gas engine.
(*e*) Steam turbine.
(*f*) Old apparatus.

[1] Report of Committee on Prime Movers, N.E.L.A., 1914.

A summary of the deductions regarding each of these methods of meeting peak loads follows.

(*a*) Storage Batteries.—Fixed charges excessive on a battery capable of discharging at maximum rate for 2 hr.

(*b*) Purchased Power.—Heavy fixed charges per kilowatt of maximum demand ($15 to $25 per kilowatt) plus charge per kilowatt-hour of actual service.

(*c*) Hydroelectric Power.—If transmitted any distance shows heavy fixed charges.

(*d*) Gas Engine.—Low fuel cost while idle, but heavy fixed charges.

(*e*) Steam Turbines.—Good because of low first cost.

(*f*) Old Apparatus.—Best because the interest on first cost must be paid or written off as depreciation. As apparatus becomes obsolete, it is kept a few years as peak apparatus.

As a matter of fact any unit, no matter how uneconomical, is good peak apparatus since it is only used a few hours per year and the cheapest fixed charge determines what is best for that small amount of service.

PROBLEMS

59. Given a 1,200-kw. condensing railroad unit—direct-connected type—comprising the following apparatus:

Boilers.—Two batteries of two boilers each, three corresponding to the full rated capacity of the plant, one for reserve.

Engines.—One unit—cross, compound gridiron valve type—developing rated capacity with 160 lb. gage pressure and 24-in. effective vacuum.

Condensers.—Surface type, for 26.6-in. vacuum in shell, with circulating water at 70°F.; ratio, water to steam 40 to 1; head on circulating pump, 25 ft.

Air and Circulating Pumps.—Horizontal, single, steam-driven, direct-acting.

Feed and Oil Pumps.—Horizontal, duplex, direct-acting, steam-driven.

Heater.—Open type, steam from feed, air and circulating pumps.

Fuel.—Cal. crude oil, 18,500 B.t.u. per pound, 336 lb. per barrel.

Required:

(*a*) The full-load test economy of the plant in terms of kilowatt-hours output at the switchboard per barrel of oil burned, with constant full load.

(*b*) The test economy of the plant in terms of kilowatt-hours output at the switchboard per barrel of oil burned, including all standby losses, if demand on the plant is as follows:

6:00 a.m.–8:00 a.m.	1,200 kw.
8:00 a.m.–9:30 a.m.	850 kw.
9:30 a.m.–3:00 p.m.	425 kw.
3:00 p.m.–4:30 p.m.	850 kw.
4:30 p.m.–6:30 p.m.	1,200 kw.
6:30 p.m.–midnight	425 kw.
Midnight–6:00 a.m.	250 kw.

60. A power plant consists of:

1. A 200-hp. simple non-condensing steam engine with direct-connected 125-kw. D.C. generator. This unit cost complete, erected, $5,000.

2. Two B. & W. boilers of 2,550 sq. ft. of heating surface each, costing complete with chain-grate stokers, erected, $9,300; two feed pumps at $200 each and feed-water heater costing $500.

3. A brick stack costing $4,250.

4. Boiler and engine houses and coal pockets costing $15,400.

5. Incidental erecting costs $3,355.

The output of the plant during the last year, operating 10 hr. per day for 308 days, was 346,000 kw.-hr.

The plant was also called upon to supply heat 10 hr. per day for a total of 5,400 sq. ft. of radiation. The exhaust steam was used for heating.

An electric company makes a proposition to supply the power at 2.5 cts. per kw.-hr., from its central station, the heating still to be done by the private plant.

To take advantage of the central station power, it will be necessary to change from the D.C. to the A.C. system. If this is done the new A.C. motors will cost $11,500.

About $3,675 (40 per cent. of initial cost) can be realized on the D.C. motors. The D.C. wiring cost $10,200. The additional wiring to change to A.C. system will amount to $1,870.

The labor required during the past year consisted of one engineer at $1,100, one fireman at $780, one fireman 6 months at $300.

Will it pay to purchase power and simply use the isolated plant for heating, or is it best to turn down the proposition of the electric company?

A. Cost of Operating Private Plant.

Determine cost of:

1. Water for power and exhaust steam heating.
2. Water for live steam heating, if any.
3. Coal for power and exhaust steam heating.
4. Coal for live steam heating, if any.
5. Oil, waste, etc.
6. Attendance.
7. Fixed charges.

B. Cost with Purchased Power.

Determine cost of:

1. Power purchased.
2. Water for heating.
3. Coal for heating.
4. Oil, waste, etc.
5. Attendance.
6. Fixed charges.

61. It is probable that in 2 years the demand upon the plant in problem 60 will double the kilowatt-hour output. At the same time the manufacturing business will require heat 10 months, 24 hr. per day, the radiating surface amounting to 32,500 sq. ft. Will it pay to purchase power and run the isolated heating plant or will it be better to operate the isolated plant for both power and heat, adding a 500-hp. engine with 325-kw. D.C. direct-connected generator? The cost of this unit erected, complete, will be $15,000. The motor installations of problem 60 will be ample to take care of the extra demand.

CHAPTER XIV

COST OF POWER

The *Iron Age*, July 27, 1911, gives the following figures for the cost per kilowatt-hour for a 2,000-hp. plant located near the colliery mouth using slack coal at 30 cts. per ton at the mine and 90 cts. per ton delivered.

The plant furnished power for the railway shops and all uses of power were reduced to the kilowatt-hour basis.

Output for the year	2,472,513 units (kw.-hr.)
Coal fired	15,414 tons
Coal per kilowatt-hour	13.95 lb.
Coal per kilowatt-hour	0.56 cts.
Average yearly cost per kilowatt-hour (station cost)	0.877 cts.
The steam plant cost	$250,000
Fixed charges per kilowatt-hour	1.215 cts.
Total cost	2.092 cts.

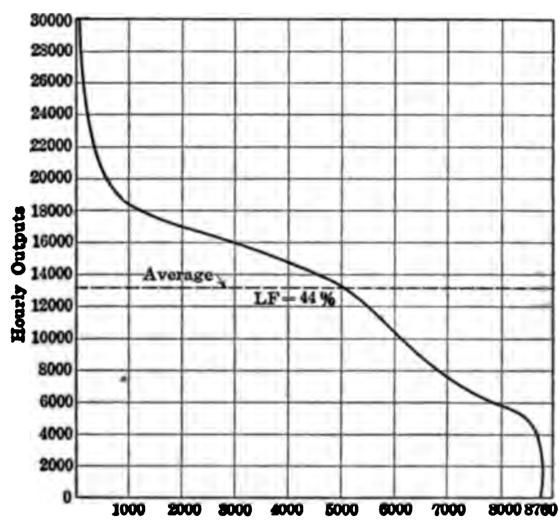


FIG. 178.—Load duration curve of Rochester Railway Light & Power Co.

The *Engineering Record*, Jan. 21, 1911, gives the following figures for the station of the Edison Electric Ill. Co. of Brocton, Mass.:

Year	Output, kw.-hr.	Use factor, per cent.	Load factor, per cent.	Cost of coal, cents	Labor, cents	Coal per ton, dollars	
1907...	2,831,000	21.0	26.5	1.23	0.34	4.91	Engine plant, 1,700 kw.
1909....	5,868,000	22.3	28.6	0.62	0.29	4.45	Turbine station, 3,000 kw.
1910....	8,079,000	30.7	33.1	0.56	0.29	4.27	Turbine station, 3,000 kw.

Unfortunately in America but few good steam-plant station costs a published on a comparable basis. In England much more publicity given to these figures and *Lighting* and a few other papers publish

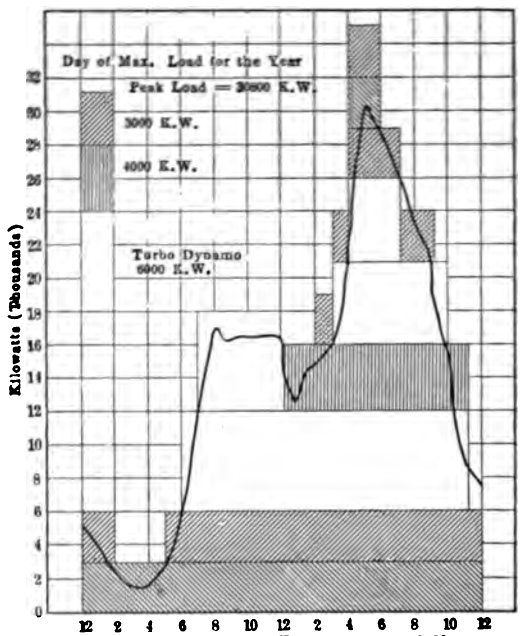


FIG. 179.—Load curve of large central station. (Josse.)

every issue a table of the costs, outputs, and other data for most of lighting and traction plants. These figures have now been published many years and are of great interest. The following tables have b taken from these reports.

Stahl und Eisen, Dec. 21, 1911, published a table giving the stat costs for 37 steam stations covering plants in Germany, Austria, Rus

BLE SHOWING COST OF POWER IN ENGLISH ELECTRIC STATIONS (*Municipally owned*)

lectric supply stations	Year	Yearly load, kw.-hr.	Coal and other fuel	Oil, waste, water and stores	Wages of work-men	Repairs and main-tenance	Total cost	Max. load on feeders	Load factor	Plant capac-ity at end of year
Marylebone.....	1910	10,776,459	0.586	0.0406	0.244	0.426	1.300	7,824	15.72	12,000
rdeen..........	1910	5,436,065	0.630	0.0203	0.183	0.183	1.016	3,218	19.28	4,649
ningham........	1911	32,866,835	0.366	0.0203	0.162	0.245	0.793	15,553	24.12	22,040
ton............	1911	11,156,084	0.550	0.0408	0.163	0.184	0.938	5,019	25.37	7,600
dford..........	1910	18,737,857	0.325	0.0610	0.142	0.245	0.773	7,922	27.00	8,180
ghton..........	1911	10,285,680	0.750	0.0460	0.265	0.428	1.489	5,140	22.84	7,200
nburgh.........	1910	15,309,493	0.530	0.0407	0.102	0.366	1.039	11,424	15.30	15,217
sgow...........	1910	36,479,243	0.428	0.0407	0.184	0.265	0.917	21,719	19.17	37,478
ds.............	1911	14,372,765	0.305	0.0203	0.122	0.225	0.672	7,980	20.56	15,740
rpool..........	1910	36,089,627	0.429	0.0407	0.184	0.143	0.797	18,071	22.80	37,000
nchester.......	1911	83,308,848	0.407	0.0203	0.203	0.184	0.814	37,520	25.35	47,301
tingham........	1911	11,944,527	0.731	0.1420	0.285	0.285	1.443	6,316	21.69	10,850
ord............	1910	14,719,170	0.530	0.0610	0.122	0.203	0.916	6,707	25.05	7,000
field..........	1910	10,317,933	0.325	0 0203	0.184	0.325	0.854	6,870	17.14	11,400
ybridge........	1911	13,295,341	0.325	0.0202	0.081	0.061	0.487	5,300	28.64	8,003
derland........	1911	10,208,493	0.345	0.0202	0.122	0.265	0.752	5,235	22.26	9,590
t Ham..........	1911	22,690,266	0.470	0.0203	0.122	0.203	0.815	8,123	31.89	11,400

BLE SHOWING COST OF POWER IN ENGLISH ELECTRIC STATIONS (*Privately owned*)

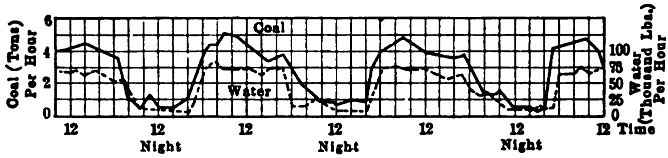
ectric supply stations	Year	Yearly load, kw.-hr.	Coal and other fuel	Oil, waste, water and stores	Wages of work-men	Repairs and main-tenance	Total cost	Max. load on feeders	Load factor	Plant capac-ity at end of year
tral...........	1909	17,282,370	0.547	0.061	0.142	0.162	0.912	8,616	22.90	12,830
ring Cross......	1910	25,733,222	0.770	0.040	0.284	0.426	1.520	13,231	22.20	21,440
lsea............	1910	4,144,936	1.112	0.081	0.506	0.588	2.287	2,769	17.09	3,500
of London.....	1910	25,183,380	0.608	0.020	0.243	0.304	1.175	17,767	16.18	25,000
nty of London	1910	16,985,687	0.648	0.040	0.182	0.507	1.377	10,350	18.73	16,500
don............	1909	10,308,537	0.709	0.061	0.284	0.386	1.440	7,756	15.17	17,250
tropolitan.......	1910	12,287,674	0.770	0.040	0.446	0.507	1.763	7,800	17.98	18,500
stminster........	1909	18,546,815	1.032	0.101	0.304	0.467	1.904	10,169	20.82	17,225
wcastle Dist.....	1909	10,479,253	0.405	0.081	0.162	0.162	0.810	5,530	21.63	9,000
ckney...........	1910	4,785,053	0.528	0.041	0.162	0.162	0.893	2,857	19.12	4,800
hton on Lyne....	1910	2,838,295	0.508	0.081	0.243	0.446	1.278	1,391	23.29	1,760
ry...............	1911	4,014,247	0.406	0.041	0.142	0.182	0.771	2,022	22.66	2,000
mbuslang........	1910	112,527	0.101	0.264	0.426	0.487	1.278	120	10.70	200
ventry..........	1911	7,443,937	0.324	0.020	0.142	0.122	0.608	4,769	19.01	6,600
arlington........	1911	2,192,036	0.386	0.041	0.203	0.162	0.792	1,235	20.26	1,970
ewsbury..........	1910	1,209,675	0.527	0.081	0.406	0.609	1.623	960	14.38	1,050
oughborough......	1911	531,886	0.345	0.041	0.365	0.487	1.238	355	17.10	600
Motherwell........	1910	2,444,944	0.406	0.081	0.182	0.122	0.791	1,280	21.80	1,910
Wolverhampton....	1910	8,289,720	0.426	0.041	0.122	0.304	0.893	4,181	22.64	5,230
Prescot...........	1910	4,560,397	0.345	0.041	0.162	0.162	0.710	1,803	28.87	2,260

TABLE SHOWING COST OF POWER IN ENGLISH ELECTRIC TRAMWAY STATIONS

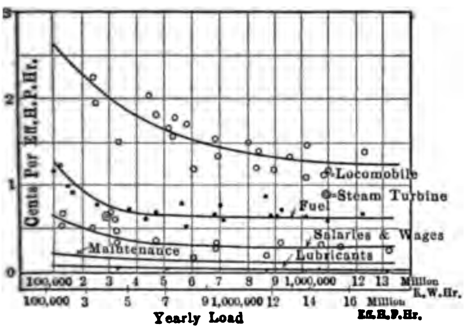
Electric railway stations	Year	Yearly load, kw.-hr.	Coal and other fuel	Oil, waste, water and stores	Wages of workmen	Repairs and main-tenance	Total cost
blin............	1910	11,771,939	0.510	0.041	0.203	0.102	0.858
sgow............	1910	26,860,126	0.245	0.041	0.203	0.162	0.651
s...............	1910	14,793,687	0.470	0.041	0.143	0.041	0.695
stle-on-Tyne.....	1910	8,213,651	0.305	0.020	0.184	0.203	0.712
d...............	1910	13,960,753	0.385	0.020	0.142	0.162	0.712
....	1910	4,047,580	0.590	0.041	0.163	0.061	0.860
eld.............	1911	4,751,903	0.366	0.020	0.184	0.470	1.040
....	1910	5,799,492	0.366	0.061	0.203	0.0815	0.710
....	1910	1,596,645	0.407	0.020	0.142	0.245	0.815

Operating Results of 37 Steam Central Stations, S & E, Dec. 21, 1911)

	Stations	Yearly load in million kw.-hr.	Kw.-hr. produced in per cent. of capacity of plant ÷ 8,760 hr.	Coal cost: Aver. cost per ton (2,240 lb.)	Coal cost: Aver. cost per 10,000 B.t.u.	B.t.u. per pound	Pounds of coal per kw.-hr.	Costs per kw.-hr. (cents): Fuel	Lubricants	Wages	Maintenance and supplies	Incidentals	Total	B.t.u. per kw.-hr.
		Mill. kw.-hr.	Per cent.	Dollars	Cents	B.t.u.	Pounds	Cents	Cents	Cents	Cents	Cents	Cents	B.t.u.
1	Amsterdam E.W.	19.88	14.2	2.82	.....	11,600	....	0.538	0.017	0.284	0.330	0.162	1.40	
2	Barmen	11.60	16.4	3.61	0.120	13,050	4.16	0.905	0.007	0.362	0.235	0.079	1.58	52,000
3	Berlin (a)	215.60	19.2	3.85	0.150	11,400	2.60	.....	.....	.....	.....	.....	....	29,600
4	Bremen	12.90	19.1	4.50	0.144	13,700	3.29	0.785	0.017	0.625	0.395	0.340	2.16	45,100
5	Breslau	14.55	19.6	2.83	0.102	12,600	3.31	0.526	0.029	0.598	0.265	0.117	1.53	41,800
6	Budapest A. EG	19.17	20.6	2.84	0.138	9,000	4.32	.....	.....	.....	.....	.....	....	38,900
7	Budapest Ung. E. A. G.	12.40	17.8	2.83	0.138	9,000	6.90	1.480	0.069	1.220	.....	2.480	5.18	62,000
8	Chemnitz	10.44	16.0	3.81	0.174	9,900	3.90	0.835	0.017	0.370	0.064	0.107	1.40	38,600
9	Koln a Rh.	33.36	25.4	3.72	0.120	13,500	2.79	0.540	0.262	0.285	.....	.....	1.20	37,700
10	Dortmund St. E. W.	18.52	17.6	3.26	0.114	13,050	2.89	0.746	0.014	0.295	0.086	0.198	1.34	37,800
11	Dresden Kraftwerk	23.49	27.0	2.59	0.150	7,600	6.64	0.762	0.038	0.320	0.248	0.112	1.48	50,500
12	Düsseldorf	16.86	14.1	3.16	.....	......	.....	0.627	0.029	0.450	0.036	0.238	1.38	
13	Elberfeld	14.70	22.3	3.34	0.120	12,600	2.86	0.495	0.012	0.362	0.205	0.005	1.08	36,000
14	Essen (Ruhr), R. W. E.	86.90	44.2	.....	.....	13,500	.....	.....	0.007	0.129	0.107	0.126	0.93	
15	Frankfurt a. M., W. I.	34.03	....	4.33	.....	13,500	2.38	0.541	0.002	0.450	0.138	0.088	1.22	32,200
16	Genua	14.38	21.3	7.30	0.234	13,500	3.22	.....	.....	.....	.....	.....	....	43,500
17	Hagen i.W., E. W. Mark (a)	13.33	25.4	3.28	0.114	13,150	2.75	0.452	0.007	0.136	0.083	0.072	0.75	36,200
18	Hamburg, Zollv. Niederl.	16.39	33.4	4.28	0.138	13,500	2.67	0.481	0.017	0.560	0.095	1.320	2.55	36,100
19	Hamburg Bille	17.37	15.2	3.97	0.138	12,600	2.86	0.705	0.014	0.615	0.079	1.820	3.23	36,100
20	Kiew	13.42	20.4	6.55	0.210	13,500	.....	1.330	0.004	0.615	0.340	0.770	3.10	
			•	18.60	0.400	20,700								
21	Copenhagen	25.60	17.5	3.65	0.132	11,900	2.79	0.567	0.017	0.640	0.325	0.140	1.69	33,200
22	Leipzig	12.18	....	1.13	0.108	4,340	11.55	.....	.....	.....	.....	.....	....	50,800
23	Lodz (a)	14.60	20.0	4.60	0.192	10,800	3.00	0.712	0.017	0.550	0.090	0.550	1.92	32,400
24	Magdeburg	12.93	18.5	1.42	0.120	5,250	10.10	0.818	0.014	0.470	0.143	0.166	1.61	53,100
25	Mannheim St. E. W.	11.91	20.6	4.12	0.138	13,350	3.00	0.725	0.012	0.343	0.019	0.119	1.40	40,000
26	Moscow	25.83	16.0	8.00	0.252	14,000								
			•	13.95	0.324	19,000								
27	Mulhausen i. E.	16.72	16.6	4.72	0.156	13,500	2.76	0.689	0.010	0.219	0.079	0.158	1.15	37,200
28	Oberschl. Ind. Bez.	82.81	24.4	1.28	0.054	10,800	3.20	0.209	0.007	0.145	0.093	0.086	0.54	34,800
29	Petersburg	19.47	19.8	5.77	0.168	15,000	2.66	.....	.....	.....	.....	.....	....	39,900
30	Rotterdam	12.05	18.1	.....	.....	......	.....	0.681	0.017	0.750	0.295	0.172	1.91	
31	Sampierdarena	19.54	15.7	6.20	0.192	12,700	2.76	.....	.....	.....	.....	.....	....	35,100
32	Saarbrücken-Luisental (a)	18.96	30.0	2.87	0.114	11,300	2.33	0.031	0.002	0.107	0.002	.....	0.325	26,400
						18,500								
33	Schönenberg	24.24	26.0	4.86	0.156	12,200	.....	0.723	0.017	0.489	0.148	0.690	2.06	
						11,700								
34	Stockholm	19.38	10.6	2.64	0.114	10,200	3.08	.....	.....	.....	.....	.....	....	31,400
				6.28	0.234	12,600	5.31	0.591	0.017	0.635	0.356	0.710	2.30	67,000
35	Strassburg i. E.	20.86	25.0	4.86	0.162	13,300	2.48	0.750	0.064	0.510		0.630	1.95	33,000
36	Waldenburg i Schl.	20.70	30.2	1.95	0.066	10,300								
[illegible]	[illegible]	[illegible]	[illegible]	0.49	[illegible]	5,400	[illegible]	[illegible]						



G. 180.—Curve showing water and coal consumption of a large heating station. (Josse.)



G. 181.—Operating cost steam power plants up to 1000 kw. capacity. (Josse.)

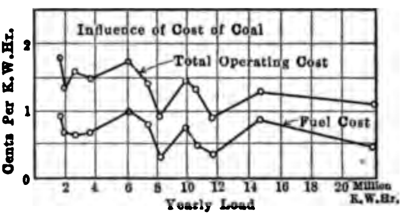


FIG. 182.—Influence of cost of fuel on total operating cost. (After Josse.)

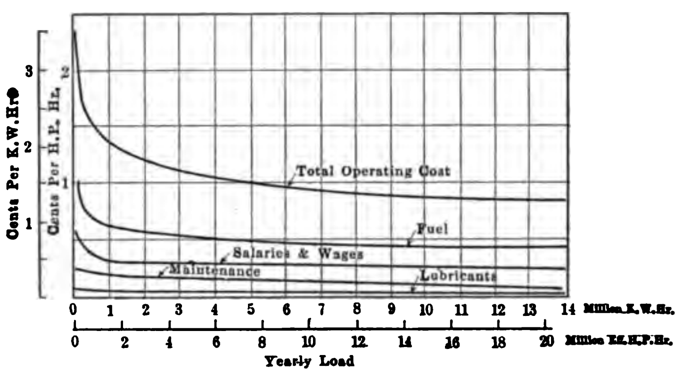


FIG. 183.—Operating costs steam power plants, reciprocating engines and steam turbines. (Josse.)

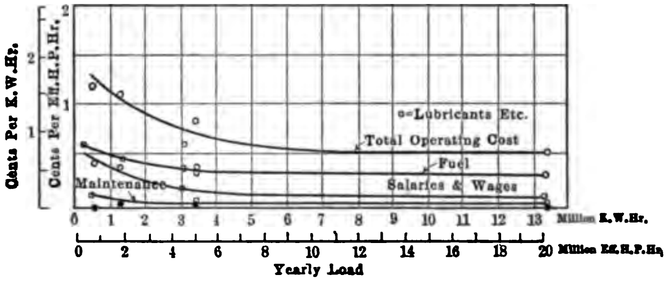


FIG. 184.—Operating costs larger sizes of steam turbine plants. (Josse.)

Sweden, Denmark and Holland. These figures are extremely int and will repay careful study.

In his "Neuere Kraftanlangen," Professor Josse of Charlot has collected figures from practically all the larger plants in G and both his figures and curves are of great value. The book a tains similar tabulations for gas and oil engines. Figs. 180 to reproduced from his book.

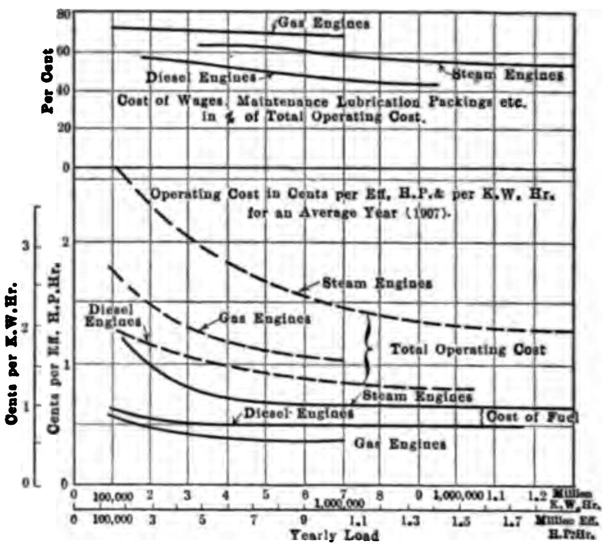


FIG. 185.—Curves of direct operating expenses. (Josse.)

The following tables of station costs for stoker-fired static hand-fired stations have also been calculated in percentages fo comparison.

The following figures have been published for the station cost bine stations with mechanical stoking:

Output	Boston Edison		Vitry (Paris), 43,000,000 kw.-hr.		Brussells Tramways, 22,000,000 kw.-hr.		Greenwi 67,000,0
	Cents	Per cent.	Cents	Per cent.	Cents	Per cent.	Cents
Coal............	0.394	74.0	0.46	75.6	0.389	80.3	0.4731
Labor...........	0.0897	16.8	0.104	17.1	0.0662	13.66	0.0718
Oil and waste....	0.0123	2.3	0.004	0.7	0.0125	2.58	0.0159
Water...........	0.0079	1.5					
Maintenance.....	0.0288	5.4	0.04	6.6	0.0167	3.45	0.0366
Total.........	0.5327	100.0	0.608	100.0	0.4844	99.99	0.5974

The following figures have been published for hand-fired stations with both engine and turbine machinery:

Output	Engine station Cambridge, 7,344,392 kw.-hr.		Engine station Boston Edison kw.hr.		Turbine station Hudson Co., 20,000,000 kw.-hr.		Engine station, 886,600 kw.-hr.		Engine station, 5,952,936 kw.-hr.	
	Cents	Per cent.	Cents	Per cent.	Cents	Per cent.	Cents	Per cent.	Cents	Per cent.
Coal	0.46	47	0.3750	54.5	0.195	47.2	1.22	57.0	0.575	54.5
Labor	0.30	31	0.2123	31.0	0.173	41.9	0.705	33.0	0.341	32.3
Oil and waste / Water	0.22	22	0.0279	4.0	......	......	0.031	1.4	0.016	1.5
			0.0308	4.5	0.02	4.8	0.025	1.1	0.052	4.9
Maintenance	....	....	0.0406	6.0	0.025	6.1	0.160	7.5	0.070	6.6
Total	0.98	100	0.6866	100.0	0.413	100.0	2.141	100.0	1.054	99.8

It will be noticed that with stoker-fired stations the fuel cost varies from 74 to 80.3 per cent. of the station cost. These figures date from 1908 to 1911 and are all from comparatively medium-sized plants, the plant of the Boston Edison Co. being the largest. With larger sized plants and 1915 conditions this percentage will probably be nearer 90 per cent. and for stoker-fired plants in general with turbines for prime movers it is safe to take the fuel cost as 80 to 85 per cent. of the station cost.

In hand-fired engine stations the fuel cost varies from 47 to 57 per cent. of the station cost. In more modern turbine plants the percentage may increase to 65 or 70 per cent. with size and good operating. Sixty per cent. is a fair figure for estimates for a turbine plant and 50 to 55 per cent. for an engine plant.

Williams and Tweedy give the inserted table of kilowatt-hour cost of electric power in steam-driven generating plants of various sizes.

As a general rule the better the plant, the larger the plant and the better the operating, the higher will be the percentage of fuel cost to the total station cost.

Standards of Good Operation.—While the station cost may be used as a criterion of good operation, the cost of coal and its quality, the prevailing rates for labor and a number of smaller factors must be known before two station costs can be compared with any degree of certainty. For stations under the same management the station cost is a good criterion but even here there may be defects in the design of the plant which will prevent economical operation, or a low use factor which will greatly increase the unit costs.

The coal consumption, pounds per horsepower-hour or per kilowatt-hour, has been used as a criterion and is a somewhat better one than station cost since only two accounts must be kept to determine it accurately and these two accounts are always kept by both large and small plants. The water rate per kilowatt-hour has also been used but is very rarely obtained with accuracy, requiring the installation of many very accurate water meters in the ordinary large plants. The ordinary city water meters can be used for the make up if calibrated very frequently.

Station	Output, kw.-hr.	Lb., coal per kw.-hr.	B.t.u. per lb. of coal	B.t.u. per kw.-hr.	L.F., per cent.
Carville, 6 mos. ending June 30, 1905.	14,604,800	3.13	11,000	34,430	37.0
Glasgow Corporation, 1905	20,558,500	4.50	10,500	47,250	17.4
Manchester (Stuart Street), 1905	28,189,455	3.57	13,500	48,200	36.3
Powell-Duffryn Steam Coal Co., 1905	4,500,000	3.75	13,000	48,750	37.0
City and South London Co., 1905	6,644,131	4.41	11,500	50,720	35.0
Charing Cross Co. Bow Stat., 1905	12,174,104	3.64	14,000	50,960	13.7
Charing Cross Co. Bow Stat., 1904	10,340,657	3.43	15,000	51,450	13.4
Sheffield Neepsend, 1905	3,499,428	4.04	13,000	52,520	13.4
Metropolitan East Side Co., 1905	22,711,000	4.64	11,800	54,750	22.0
Central Co., 1905	7,102,960	4.20	14,000	58,800	12.5
County of London Co., 1905	11,350,000	5.50	11,000	60,500	18.9
Salford, 1905	10,666,001	4.37	14,320	62,580	28.0
Leeds, 1906	8,436,817	7.15	11,000	78,650	14.5
St. James & Pall Mall Co., 1905	6,654,217	5.54	14,200	78,670	18.6
London Elec. Co., 1905	14,235,423	4.60	12,000	55,200	25.0
Bradford, 1905	14,723,356	4.12	13,000	53,560	28.0
Westminster Co., 1905	11,616,914	4.96	14,394	71,390	27.0
Berlin, 1905	141,059,129	2.38	12,368	29,440	30.4
Berlin, 1904	113,389,947	3.10	12,576	38,980	31.1
Vienna, 1904	45,939,840	2.70	11,938	32,230	35.2
Eberfeld, 1904	7,206,950	3.00	12,420	37,260	27.2
Hamburg Zollverein, 1904	12,914,177	3.00	13,500	40,500	38.6
Frankfort a/Main, 1904	16,431,832	3.30	13,500	44,550	29.9
Hamburg Combined, 1904	27,188,640	3.40	13,500	45,900	28.4
Cohn a/Rhine, 1904	13,126,850	3.60	12,870	46,330	37.8
Munich, 1904	12,888,991	3.70	12,765	47,230	24.2
Copenhagen, 1904	13,280,515	3.90	12,519	48,820	29.3
Charlottenburg, 1904	6,747,000	4.50	11,340	51,030	24.0
Oberschlesischer Indus., 1904	27,286,995	4.80	10,800	51,840	35.2
Dresden Power, 1905	12,528,657	6.50	8,244	53,580	30.8
Dresden Light, 1904	5,464,405	7.20	7,560	54,430	22.9
Brussells Tramways, 1907	21,913,000	2.09	12,000	25,100	40.0(?)
Buenos Ayres, 1904	32,722,381	3.00	13,500	40,500	42.0(?)
Brocton, Mass., 1907	2,831,000	5.62	14,000	78,800	26.5
Brocton, Mass., 1909	5,868,000	3.21	14,000	45,000	26.6
Brocton, Mass., 1910	8,079,000	3.41	14,000	47,700	31.3
Redondo, Cal., 15-day test			.....	24,438	60.0
Redondo, Cal., 16 mos., 1908–09			.....	26,200	55.0

Station water rates as low as 14 lb. per kilowatt-hour over a period of one year have been reported from stations using electric auxiliaries but rates from 17 to 20 lb. are very good and rates from 25 to 50 lb. or higher are not uncommon. Any figure between 2 and 3 lb. of coal per kilowatt-hour is good practice and there are very few reported figures below 2 lb.

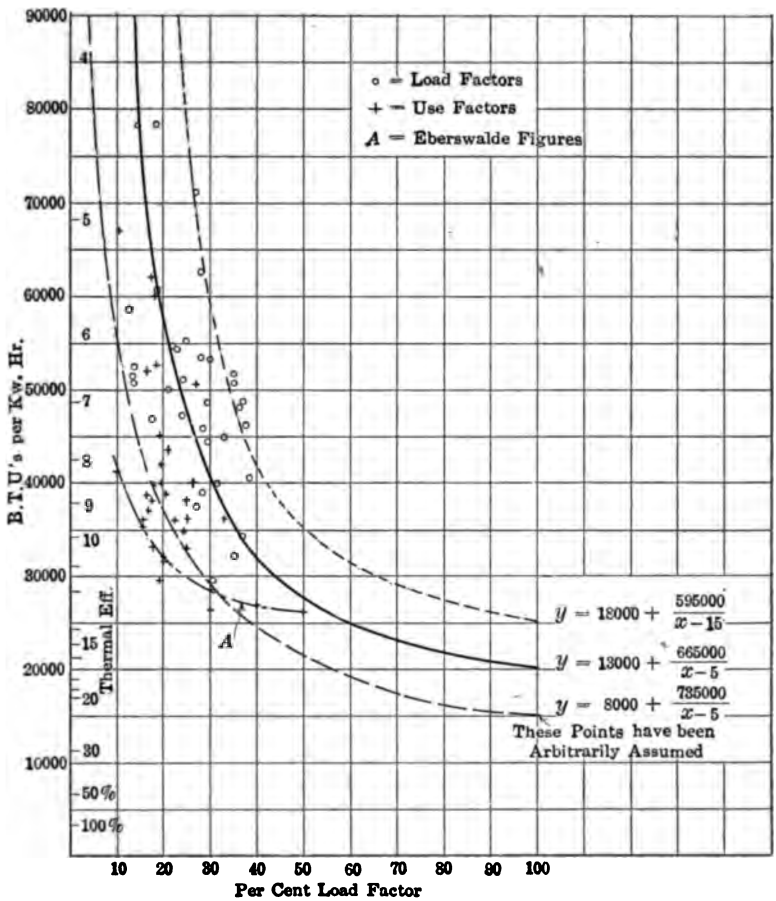


FIG. 186.—Relation between load factor and thermal units in fuel.

per kilowatt-hour. Five pounds of coal per kilowatt-hour with very low load factor is good.

Perhaps the best method of stating station economy is to give the B.t.u.'s in the coal per kilowatt-hour. This eliminates price and quality of the coal and if the load factor is stated, say 28,000 B.t.u's. per kilowatt-hour at 30 per cent. load factor, we have a very good criterion of both the design and operation of the station. This was recognized as far back as

1905 by Patchell who, in a paper in the *Proceedings* Institute of Electrical Engineers (London), vol. 39, 1905, gives the figures for a large number of stations. The preceding table is taken largely from Patchell's paper with other figures that have been published since that time.

These values of B.t.u's. per kilowatt-hour have been plotted as ordinates against load factors as abscissas in Fig. 186. The points obviously cannot lie on any one curve since the stations are of all sorts and conditions, and the operators are of various degrees of skill, but the points all lie in a field of elongated curved shape and the outside limits may be sketched in with substantial accuracy. In this field all reasonable values will appear. The median curve through the length of this field corresponds to the equation $(X - 5)(y - 13{,}000) = 665{,}000$.

It will be observed that good records lie to the left of the curve and poor records to the right.

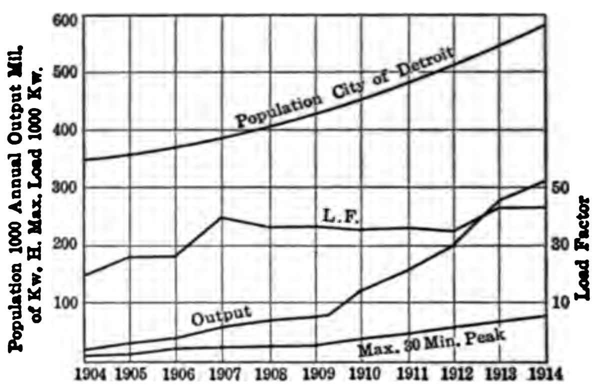


FIG. 187.—Detroit Edison. Co., development curves. (See Hirschfeld's paper, A.S.M.E., Dec., 1916.)

This measure of efficiency, B.t.u.'s per kilowatt-hour, may be turned into thermal efficiency by dividing it into 3,410 as shown by the second scale of ordinates appearing on the figure. Of the two efficiencies B.t.u's. per kilowatt-hour appears to be the better to use from an operating standpoint since the coal is generally bought on a B.t.u. basis and the cost of 1,000 B.t.u's. in the coal is usually well known.

It is well to be cautious in accepting figures published in the technical press relating to station economy. As they usually appear they are mainly defective, that is, conditions are not wholly stated. A particularly good station cost may be quoted leaving out load factor and cost of coal. Sometimes the coal cost is given as lb. per kilowatt and the fact that it is kilowatt-hours generated and not kilowatt-hours leaving the busbars is not stated. As much as 10 to 20 per cent. may be used in the station.

The loss after leaving the feeder switches should be charged to distribution, but all losses and used current in the station are not part of output. Load factors should always be yearly. It would be still better if use factors were used but this is not general practice as yet.

Many cost figures leave out repairs or maintenance and make no note of it. One company bought about one-third of its output from a water-power company but the entire operation was charged to the steam and no note made of the water in the publication. In a great many cost figures the time over which they are taken is not stated and many excellent figures of days or weeks run have been published without qualification. As no repairs were made in this time nor oil purchased, these items do not appear.

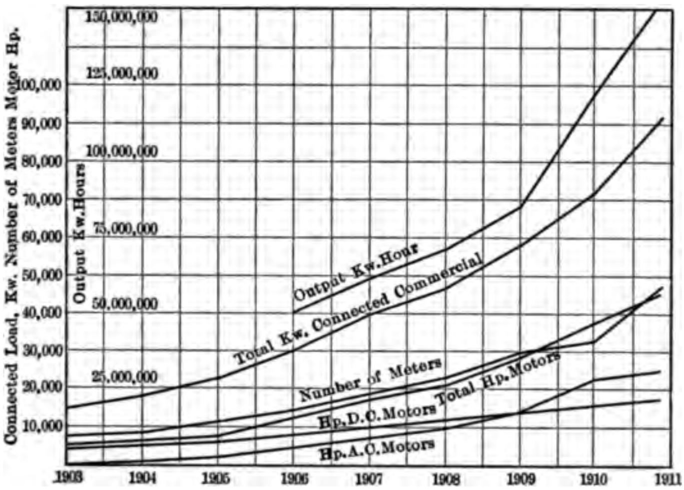


FIG. 188.—Development curve, Detroit Edison Co.

In most of these cases there is no attempt to deceive but the fault is due to the imperfect systems of bookkeeping and the reader must be able to judge of the credibility of the published figures.

In a paper before the A.I.E.E., Feb. 8, 1916, A. G. Christie gives the statistics of a number of municipal power plants in western Canada. The characteristics of the cities are reviewed and the costs and methods of financing are discussed at length. Methods of charging are also discussed. The following table has been abstracted from the paper. The fuel is mostly local lignite with some Pennsylvania and Ohio bituminous coal.

Cost of Electric Power in Western Canadian Cities

A. G. Christie

	No. boilers	No. turbines	Rated capacity	Installation cost per kw., $	Output 1,000,000 of kw.-hr.	Use factor, per cent.	Coal, lb. per kw.
Kamloops	4	2	2,700	....	2.63	11.0	6.14
Lethbridge	8	3	2,300	198.42	3.42	17.0	6.9
Moose Jaw	8	3	3,000	118.02	3.74	14.2	8.5
Saskatoon	8	3	5,950	95.80	9.72	18.6	4.6
Regina	12	7	7,600	92.70	9.32	14.0	5.85
Edmonton	16	8	10,250	190.75	21.93	24.4	5.25

	Fuel, cents	Labor, cents	Water, cents	Oil, waste and supplies, cents	Repairs and maintenance, cents	Station cost, cents
Kamloops	1.342	0.531	.....	0.041	0.004	1.918
Lethbridge	0.369	0.438	0.021	0.071	0.223	1.122
Moose Jaw	1.298	0.527	0.045	.....	.....	2.010
Saskatoon	1.367	0.326	0.029	0.045	0.130	1.897
Regina	1.412	0.440	0.009	0.046	0.087	1.994
Edmonton	0.754	0.358	0.007	0.033	0.260	1.412

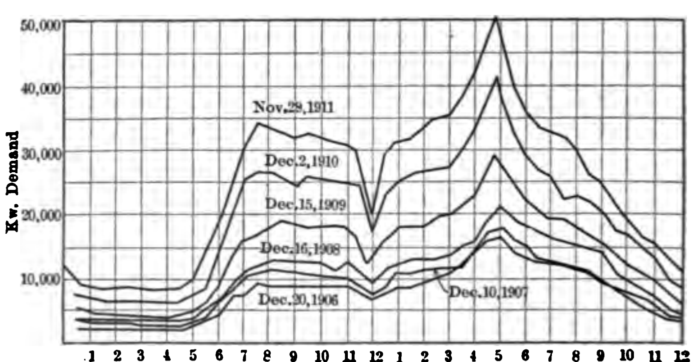


Fig. 189.—Load curve, maximum day, Detroit Edison Co.

Some very interesting figures for the large Central Station Systems in Berlin, Chicago and London are given by Klingenberg, *Proceedings* Institute Electrical Engineers (London), Dec. 4, 1913. The following table has been abstracted from his paper:

	Berlin (1911–12)	Chicago (1911)	London (1910–11)
Population	2,600,000	2,200,000	6,500,000
Power stations	6	6	64
Installed capacity, kw	137,000	221,700	298,400
Average size of station, kw	23,000	37,000	4,670
Assets	$38,950,000	$68,700,000	$132,700,000
Sinking fund	7,370,000	3,435,000	27,100,000
Real value	31,580,000	65,245,000	105,600,000
Cost per kw. installed:			
Power station	$86.45 38%	$116.00 40%	$160.80 45%
Distributing system and meters	144.00 62%	175.40 60%	193.50 55%
	230.45 100	291.40 100	354.30 100
Peak load, kw	94,600	199,300	185,500
Peak load per power station, kw	15,800	31,700	2,900
Kw.-hr. generated	274,000,000	684,000,000	405,000,000
Kw.-hr. purchased	....	32,000,000	....
Kw.-hr. sold	216,300,000	640,000,000	319,243,000
Comprising:			
Lighting, per cent	24	19	61
Power, per cent	45	12	27
Traction, per cent	31	69	12
Load factor (kw.-hr. generated) per cent	33.1	41.0	24.9
Ratio, kw.-hr. sold to kw.-hr. generated	0.79	0.894	0.788
Reserve factor	1.450	1.11	1.61
Use factor (total)	0.18	0.33	0.122
Coal, price per ton	$4.22	$1.94	$3.10
Coal consumption per kw.-hr. sold, lb.	3.015	3.54	5.21
Overall efficiency, per cent	9.7	7.6	5.0
Revenue	$8,325,000	$14,020,000	$15,130,000
Expenses	4,280,000	7,000,000	6,750,000
Profit absolute	4,045,000	7,020,000	8,380,000
Percentage of real value	12.83	10.87	7.85
Selling price, cts	3.84	2.19	4.74
Operating costs per kw.-hr. sold			
Expenses:			
Fuel	0.58 cts.	0.278[1] cts.	0.74 cts.
Oil, stores, etc	0.0945	0.0109	0.0635
Wages	0.1235	0.149	0.254
Repairs, maintenance	0.227	0.201	0.359
Rent, taxes, insurance	0.0844	0.1465	0.359
General expenses	0.2035	0.2015	0.338
Current purchased	......	0.0405	
Municipal participation	0.744	0.0657	
Total expenses	2.057	1.093	2.11
Profit gross	1.78	1.098	2.63

[1] Kw.-hr. sold includes kw.-hr. purchased; therefore the actual value is 4 to 5 per cent. higher.

The Cost of Electric Power in New York City.—The tables following, which are compiled from two articles published in *Engineering Contracting* for Apr. 6 to May 11, 1910, give unit costs of generating and distributing electricity for power and lighting in the plants of the Edison Electric Illuminating Co. of Brooklyn and the New York Edison Co. during the fiscal year ending June 30, 1907. The New York Edison Co. produces nearly 70 per cent. of the electricity used for lighting in New York City and the Brooklyn Co. is by far the most important operating in the city, next to the New York Edison Co. The combined capacities of the plants of the two companies aggregated nearly 132,000 kw.

COST OF THE BROOKLYN PLANT

	Total	Per kilowatt
Generating station buildings and land	$674,666	$29
Substation buildings and land	650,559	28
Other real estate	48,007	2
Steam plant, generating station	1,051,796	44
Electrical plant, generation station	965,824	41
Substation apparatus	507,965	21
Transmission lines	4,136,009	175
Construction, subways	230,950	10
Services	973,558	41
Line transformers	150,184	6
Meters	301,677	13
Arc lamps	242,607	10
Tools and implements	20,955	1
Office furniture and fixtures	50,131	2
Miscellaneous appliances	64,874	3
Storage batteries	182,150	8
Sundries, automobiles, etc.	74,526	3
Total	$10,326,437	$437

PRODUCTION AND SALE OF CURRENT

	Brooklyn	New York
Kilowatt-hours produced	71,769,804	299,172,431
Kilowatt-hours used in station	800,783	....
Kilowatt-hours sold	47,912,138	209,024,002
Average load factor, per cent.	34.7	31.5
Transmission losses, etc., per cent.	32.5	30.0
Average sale price, cents per kw.-hr.	6.6	6.49

COST OF POWER GENERATION AND DISTRIBUTION

Production Expense

	Brooklyn		New York	
	Total	Cents per kw.-hr.	Total	Cents per kw.-hr.
Salaries	$22,510	0.031	$35,908	0.012
Labor	136,116	0.190	508,778	0.171
Fuel	285,654	0.400	1,210,599	0.406
Oil, waste and sundries	17,355	0.024	54,565	0.018
Water	28,449	0.039	135,211	0.045
Repairs, building and structures	2,493	0.005	37,435	0.012
Repairs, motive power	39,622	0.056	211,175	0.070
Repairs, electrical apparatus	2,296	0.003	22,634	0.007
Station expense	2,330	0.003	24,893	0.008
Purchased power, electric	28,575	0.039	75,220	0.025
Total production expense	$565,400	0.788	$2,316,418	0.774

Distribution Expense

Salaries	$7,687	0.011	$64,146	0.021
Substation labor and expense	98,607	0.138	148,499	0.050
Rental of conduits, etc.	117,162	0.163	1,014,081	0.339
Incandescent lamp renewals	110,199	0.153	528,213	0.147
Wiring and jobbing	....	.....	139,713	0.047
Repairs, poles and lines	22,918	0.032	21,716	0.007
Repairs, subways and cables	30,957	0.043	56,856	0.019
Repairs, meters	4,872	0.007	221,894	0.074
Repairs, transformers	187	.....	....	.....
Repairs, substations	....	.....	202,611	0.068
Repairs and expense, commercial lamps	89,711	0.125	12,872	0.004
Repairs and expense, street lamps	....	.....	68,096	0.023
Total distribution expense	$482,300	0.672	$2,388,698	0.799

General Expense

	Brooklyn		New York	
	Total	Cents per kw.-hr.	Total	Cents per kw.-hr.
Salaries of officers	$18,850	0.029	$51,241	0.017
Salaries of office staff	67,665	0.094	335,825	0.113
Collecting and reading meters	19,797	0.028	....	.....
Office expense	......	.....	294,009	0.098
Legal expense	19,009	0.027	116,630	0.039
General expense	73,256	0.101	....	.....
Advertising and soliciting	......	.....	218,850	0.073
Advertising	44,364	0.061	....	.....
Canvassing new business	82,917	0.115	....	.....
Insurance	47,884	0.067	118,393	0.040
Engineering and testing	......	.....	78,707	0.026
Leaseholds, rentals, etc	......	.....	47,656	0.016
Miscellaneous	58,178	0.080	....	.....
Total general expense	$431,920	0.602	$1,262,311	0.422

Taxes and Miscellaneous

Taxes, general	86,916	0.121	702,628	0.235
State franchise tax	61,584	0.085	....	.....
Leaseholds, rentals, etc	4,783	0.007	....	.....
Uncollectable bills	......	.....	35,464	0.011
Depreciation and contingent expense	......	.....	1,721,413	0.575
Total taxes and miscellaneous	$153,283	0.213	$2,459,505	0.821

Summary

Total production expense	$565,430	0.788	$2,316,418	0.774
Total distribution expense	482,300	0.672	2,388,698	0.799
Total general expense	431,920	0.602	1,262,311	0.422
Total taxes and miscellaneous	153,283	0.213	2,459,505	0.821
Grand total	$1,632,903	2.275	$8,425,932	2.816

Load Curves.—The accompanying load curves and records of connected load illustrate the rapid increase in generating-plant capacity which has had to be provided to keep pace with the growing demand. The recent development of Detroit as a factory city is responsible in a measure for the unusual increase of commercial load, and the resulting influx of factory employees has developed the residence service in the newer sections.

Fig. 188 records the increase in total kilowatt-hours' output, total commercial kilowatts connected, number of meters, and the total horsepower of motors, both direct-current and alternating-current, for the years 1903 to Nov. 1, 1911.

In addition to its own lighting and commercial load, the Detroit system furnishes energy for the associated Eastern Michigan Edison Co., operating in the suburban district surrounding Detroit. About one-fifth of its total generated output is purchased by the Detroit city railways for operating cars in all outlying sections of the city. Another unusual traction load taken over by the central-station company within the year is the operation of the Detroit River tunnel of the Michigan Central Railroad. The tunnel substation takes its energy supply through a 500-kw. motor-generator set, being arranged with a storage battery so that the

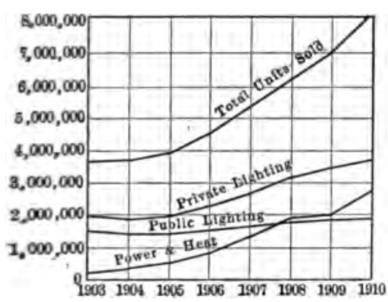


FIG. 190.—Development curve, Melbourne Elec. supply.

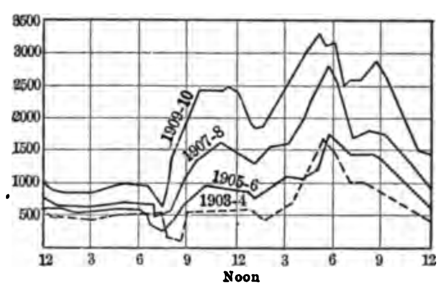


FIG. 191.—Load curve of maximum days, Melbourne Elec. supply.

peaks of demand of train acceleration are not felt by the central-station system.

Figure 187 taken from Hirshfeld's paper (A.S.M.E., December, 1916) shows the continuation of growth in its relation to population, and similar curves are given in Figs. 190, 191 and 192 for Melbourne, Australia.

Curves of daily maximum loads show the variation of demand throughout the year and should be studied very carefully as much may be learned from them.

Cost Curves at Variable Loads.—Dr. Klingenberg, in "Bau. Gr. Elek.," has shown a very convenient method for showing graphically the economy of central stations. This is possible where it is convenient to get the actual cost at two or three ratings over a sufficient period of time to reduce errors to small dimensions.

If the total cost in dollars, including fixed charges, is plotted as ordinates, against the load in kilowatts as abscissæ, the resulting curve will be a straight line intercepting the Y-axis at a fixed distance above the origin.

If the cost curve be transposed, making the Y-axis the X-axis, extend the new X-axis to the right and erect upon this a daily load curve, the hourly cost can be read off directly with suitable scales at any portion of the curve.

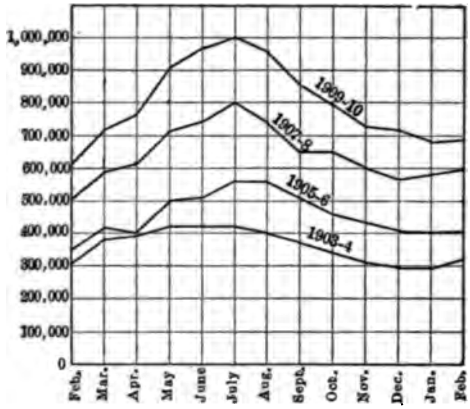


FIG. 192.—Curves of monthly outputs, Melbourne Elec. Supply.

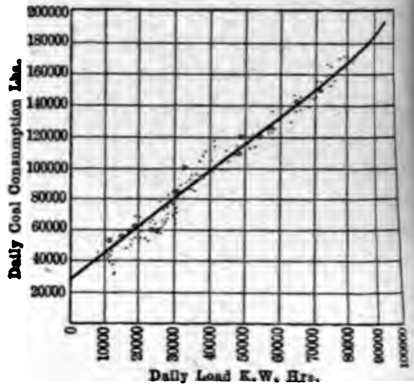


FIG. 193.—Coal consumption curve, Markische E. W.

If the intercept of the economy curve be transferred to the Y-axis below the origin and each point of the load curve be projected to the economy curve and thence transferred to the fourth angle, as in Fig. 196, an

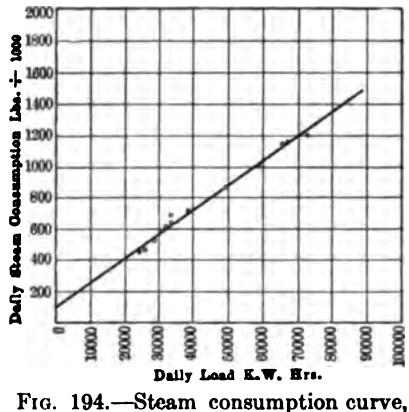


FIG. 194.—Steam consumption curve, Markische E. W.

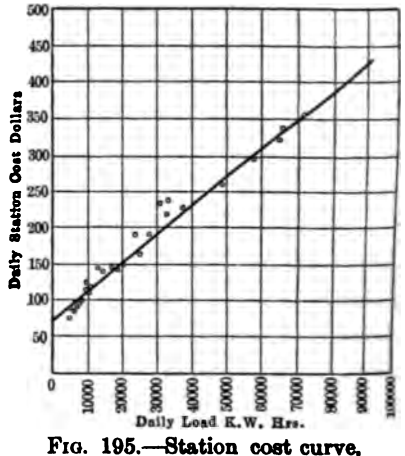


FIG. 195.—Station cost curve, Markische E. W.

area will result whose ordinates will be cost and from which the constant and variable costs can be scaled off for any hour of the day.

If yearly records are available and a plot be made with the economy curve in the second angle and the average load of each hour of the year in the first angle in the order of their magnitude, commencing e maximum hour, and these two curves be combined as before in rth angle, a yearly cost curve will be obtained. The area under rly load curve in the first angle will be the output of the station vatt-hours and the shape of the curve will show how much the ery is used and to what advantage. Similar curves may be drawn yearly steam consumption and coal consumption.

ondon Engineering, Nov. 13, 1914, R. H. Parsons has shown curves nilar character, but not so well developed, and S. A. Fletcher, in *ctric Journal*, has done similar work along this line. It should be hat the diagrams show that the smaller the use factor the greater the effect of the constant cost portion of the area under the curve. l-load efficiency of the machines, which is often the criterion, will

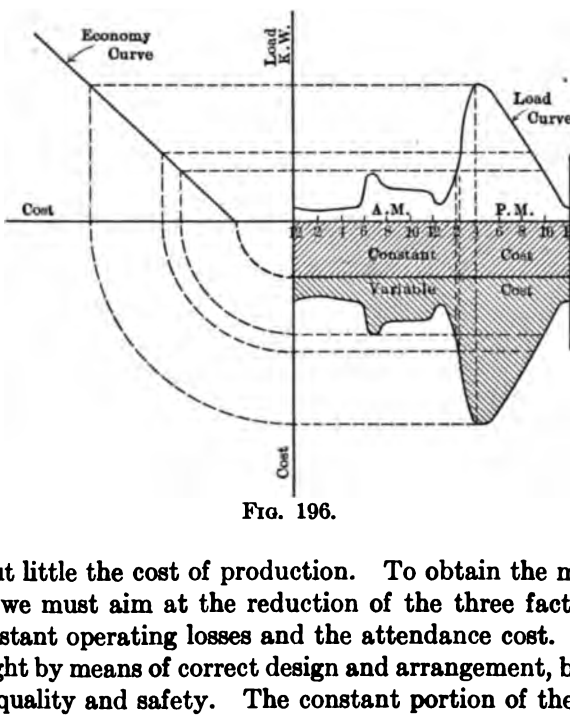


FIG. 196.

e but little the cost of production. To obtain the most econom- ults we must aim at the reduction of the three factors, invested , constant operating losses and the attendance cost. This saving sought by means of correct design and arrangement, but not at the of quality and safety. The constant portion of the load should ied by the most economical machines, reserving the cheaper equip- or the variable portion of the load. Figs. 197 and 198 give oper- esults on this basis for one of the most interesting of central s.

pages 66 to 73 of the same book, Dr. Klingenberg has estimated it and calculated the station economy for three different sizes of plants, 1,000-kw., 5,000-kw. and 20,000 kw. He has worked these figures out on the basis of various use factors from 10 per cent. to the li iting condition, 100 per cent., and his results, which are shown in Fi 199 and 200, are very interesting. These curves are analogous to t economy curve for steam engines, given so long ago by Emery.

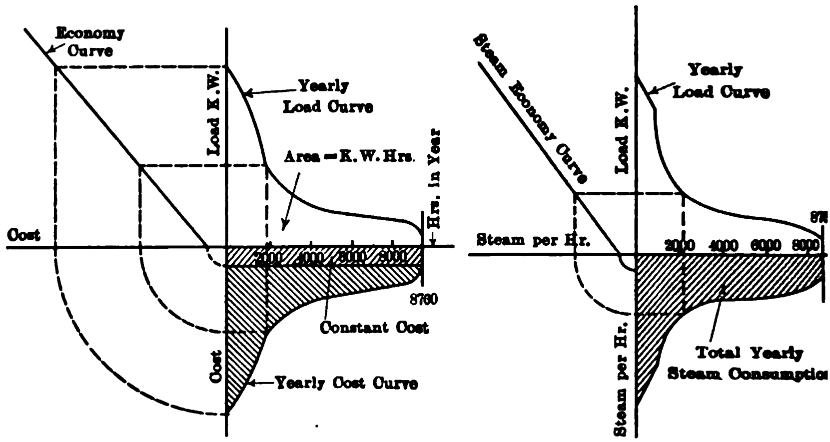


Fig. 197.

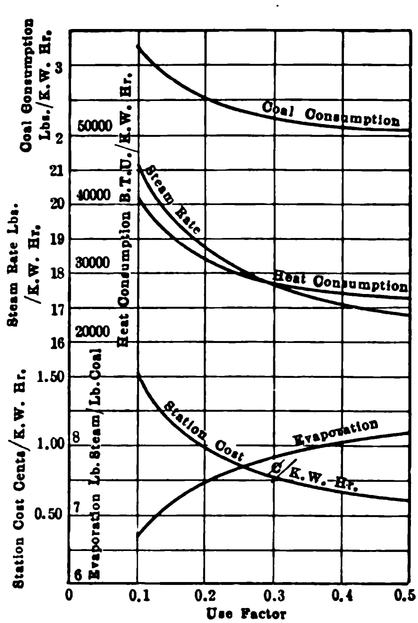


Fig. 198.—Economical results Eberswalde Station, Markische E. W.

In "Bau. Gr. Elek." he has given figures regarding the Eberswa plant of the Markischer Elektrizitatswerke. This plant, illustrated Figs. 96 and 160, had 7,400 kw. installed in 1,500-r.p.m. turbines at t time and cost $393,000 including $7,150 for land and bulkhead work. This is the cheapest central station reported and is a fine example of clever design, being well suited to the work expected of it. The cost per kilowatt installed is $53 and the fixed charges at 12 per cent. are $47,100 or $6.36 per kilowatt. Very careful records were kept for the 3-year period which the figures cover and the following table indicates first-class results following careful design and good operation.

Use factor, per cent.	Coal, lb. per kw.-hr.	B.t.u.'s per kw.-hr.	Lb. water evaporated per lb. coal	Station water rate	Operating cost	Fixed charges, 12 per cent.	Total station cost
10	3.24	41,200	6.72	21.76	0.794	0.726	1.520
20	2.495	31,700	7.47	18.66	0.590	0.363	0.953
30	2.247	28,533	7.83	17.63	0.523	0.242	0.765
40	2.133	26,950	8.05	17.11	0.490	0.181	0.671
50	2.048	26,000	8.20	16.80	0.466	0.145	0.611

Economizers are used in this station but no heaters. The auxiliaries are electrically driven with the exception of the feed pumps and condenser pumps which are driven by small steam turbines.

TABLE 1.—KLINGENBERG'S ESTIMATE

No.	Items	Power plants		
		$A = 20,000$ kw.	$B = 5,000$ kw.	$C = 1,000$ kw.
1	Boiler eff. full load including power for draft, chain grates and feed pumps, %	78	76	75
2	No-load boiler consumption in per cent. full load			
	Boiler, %	9.00	9.75	10.00
	Auxiliaries, %	1.50	1.50	1.50
	Feed pumps, %	0.50	0.50	0.50
	Total, %	11.00	11.75	12.00
3	Heat drop in turb. and cond. (185#–197° S.H.) B.t.u./#	1,240	1,240	1,240
4	Steam at full load including excit. and aux., #/kw.-hr.	12.8	14.3	16.5
5	Heat equiv. of 1 kw.-hr., B.t.u.	3,415	3,415	3,415
6	Eff. of turb. = item 5 ÷ (item 3 × item 4), %	21.5	19.1	16.6
7	No-load cons. per cent. full-load consumption, %	10.0	12.5	15.0
8	Live-steam piping surface, sq. ft.	1,940	1,400	1,076
9	Heat loss, B.t.u./sq. ft./hr.	368	368	368
10	Eff. of piping excl. of throttle loss, %	99.8	99.7	99.6
11	Power cons. (light, etc., in station) in % full load, %	0.5	0.75	1.00
12	Installation cost per kw., $	35.70	47.60	71.40
13	Water, etc., per kw. year, $	0.07	0.10	0.14
14	Wages per kw. year, $	0.70	1.00	1.43
15	Repairs per kw. year (1% station cost), $	0.36	0.48	0.73
16	Interest & renewals per kw. year (12% sta. cost) $	4.28	5.71	8.56
17	Heating value of coal, B.t.u./#	13,500	13,500	13,500
18	Cost of coal delivered to boiler per ton (2,000 #)	3.24	3.24	3.24
19	Cost per million B.t.u., $	0.12	0.12	0.12

TABLE 2

No.	Items	Heat consumption of plants at full load								
		A = 20,000 kw.			B = 5,000 kw.			C = 1,000 kw.		
		Constant part	Variable part	Total	Constant part	Variable part	Total	Constant part	Variable part	Total
20	Boiler (1–2)									
	Received	11.00	89.00	100.00	11.75	88.25	100.00	12.00	88.00	100.00
	Delivered	......	......	78.00	......	......	76.00	......	......	75.00
21	Piping (10)									
	Received	0.16	77.84	78.00	0.23	75.77	76.00	0.30	74.70	75.00
	Delivered	......	......	77.84	......	......	75.77	......	......	74.70
22	Turbine (6–7)									
	Received	7.78	70.06	77.84	9.45	66.32	75.77	11.20	63.50	74.70
	Delivered	......	......	16.70	......	......	14.50	......	......	12.40
23	Light and power									
	Received	0.08	16.62	16.70	0.11	14.39	14.50	0.12	12.28	12.40
	Delivered	......	......	16.62	......	......	14.39	......	......	12.28
	Total balance									
24	Received	19.02	80.98	100.00	21.54	78.46	100.00	23.62	76.38	100.00
	Delivered	......	......	16.62	......	......	14.39	......	......	12.28
25	Total balance per kw.-hr. full load									
	Received, B.t.u.	3,900	16,600	20,500	5,150	18,600	23,750	6,580	21,220	27,800
	Delivered, B.t.u.	......	......	3,415	......	......	3,415	......	......	3,415
26	Coal per kw.-hr., #	0.29	1.23	1.52	0.38	1.38	1.76	0.49	1.57	2.06

TABLE 3

No.	Items	Operating cost in cents per kw.-hr. at full load								
		A = 20,000 kw.			B = 5,000 kw.			C = 1,000 kw.		
		Constant part	Variable part	Total	Constant part	Variable part	Total	Constant part	Variable part	Total
27	Coal (19–25)	0.047	0.199	0.246	0.062	0.223	0.285	0.079	0.255	0.334
28	Water (13)	0.001	......	0.001	0.001	......	0.001	0.002	......	0.002
29	Wages	0.008	......	0.008	0.011	......	0.011	0.016	......	0.016
30	Repairs	0.004	......	0.004	0.005	......	0.005	0.008	......	0.008
31	Interest and renewals	0.049	......	0.049	0.065	......	0.065	0.098	......	0.098
	Total	0.109	0.199	0.308	0.144	0.223	0.367	0.203	0.255	0.458

TABLE 4

Items	Units	Symbol	Power plant		
			A 20,000 kw.	*B* 5,000 kw.	*C* 1,000 kw.
eat consumption at no load (25)	B.t.u. per kw.-hr.	a_w	3,900	5,150	6,580
dditional heat consumption (25)	B.t.u. per kw.-hr.	b_w	16,600	18,600	21,220
peratingcostwithoutcoal(32)(27)	Cents per kw.-hr.	c	0.062	0.082	0.124
ost of coal (19)................	Cents per mill B.t.u.	g	12	12	12

TABLE 5

lo.	Items Units	Power plant		
		A = 20,000 kw.	*B* = 5,000 kw.	*C* = 1,000 kw.
1	Momentary heat consumption, B.t.u./kw.-hr.	$W_t = 3{,}900\frac{1}{m} + 16{,}600$	$W_t = 5{,}150\frac{1}{m} + 18{,}600$	$W_t = 6{,}580\frac{1}{m} + 21{,}220$
4	Average yearly heat consumption, B.t.u./kw.-hr.	$W_m = 3{,}900\frac{1}{n}f + 16{,}600$	$W_m = 5{,}150\frac{1}{n}f + 18{,}600$	$W_m = 6{,}580\frac{1}{n}f + 21{,}220$
5	If (f = 1), B.t.u./kw.-hr.	$W_{m1} = 3{,}900\frac{1}{n} + 16{,}600$	$W_{m1} = 5{,}150\frac{1}{n} + 18{,}600$	$W_{m1} = 6{,}580\frac{1}{n} + 21{,}220$
6	If (f = n), B.t.u./kw.-hr.	$W_{m2} = 20{,}500$	$W_{m2} = 23{,}750$	$W_{m2} = 27{,}800$
7	Average yearly operating cost, cents/kw.-hr.......	$K = 0.062\frac{1}{n} + \frac{12 \times W_m}{1{,}000{,}000}$	$K = 0.082\frac{1}{n} + \frac{12 \times W_m}{1{,}000{,}000}$	$K = 0.124\frac{1}{n} + \frac{12 \times W_m}{1{,}000{,}000}$

TABLE 6

Use factor	Limit f = 1, n = m Equation #5						Limit f = n Equation #6		
	B.t.u. per kw.-hr.			Percentage 20,500 = 100 %			B.t.u. per kw.-hr.		
	A	B	C	A	B	C	A	B	C
1.0	20,500	23,750	27,800	100.0	86.1	74.0	20,500	23,750	27,800
0.9	20,930	24,320	28,520	98.0	84.2	72.1	20,500	23,750	27,800
0.8	21,470	25,020	29,420	95.5	81.9	69.6	20,500	23,750	27,800
0.7	22,170	24,950	30,620	91.8	79.0	67.0	20,500	23,750	27,800
0.6	23,100	27,170	32,170	88.9	75.6	63.8	20,500	23,750	27,800
0.5	24,400	28,900	34,420	84.1	71.0	59.8	20,500	23,750	27,800
0.4	26,350	31,500	37,670	77.9	65.2	54.7	20,500	23,750	27,800
0.3	29,600	35,800	43,120	69.4	57.2	47.5	20,500	23,750	27,800
0.2	36,100	44,400	54,020	56.7	46.4	37.9	20,500	23,750	27,800
0.1	55,600	70,100	86,920	36.8	29.3	23.6	20,500	23,750	27,800

TABLE 2

No.	Items	Heat consumption of plants at full load								
		A = 20,000 kw.			B = 5,000 kw.			C = 1,000 kw.		
		Constant part	Variable part	Total	Constant part	Variable part	Total	Constant part	Variable part	Total
20	Boiler (1–2)									
	Received	11.00	89.00	100.00	11.75	88.25	100.00	12.00	88.00	100.00
	Delivered	.....	.....	78.00	.....	.....	76.00	.....	.....	75.00
21	Piping (10)									
	Received	0.16	77.84	78.00	0.23	75.77	76.00	0.30	74.70	75.00
	Delivered	.....	.....	77.84	.....	.....	75.77	.....	.....	74.70
22	Turbine (6–7)									
	Received	7.78	70.06	77.84	9.45	66.32	75.77	11.20	63.50	74.70
	Delivered	.....	.....	16.70	.....	.....	14.50	.....	.....	12.40
23	Light and power									
	Received	0.08	16.62	16.70	0.11	14.39	14.50	0.12	12.28	12.40
	Delivered	.....	.....	16.62	.....	.....	14.39	.....	.....	12.28
	Total balance									
24	Received	19.02	80.98	100.00	21.54	78.46	100.00	23.62	76.38	100.00
	Delivered	.....	.....	16.62	.....	.....	14.39	.....	.....	12.38
25	Total balance per kw.-hr. full load									
	Received, B.t.u.	3,900	16,600	20,500	5,150	18,600	23,750	6,580	21,220	27,800
	Delivered, B.t.u.	.....	.....	3,415	.....	.....	3,415	.....	.....	3,415
26	Coal per kw.-hr., #	0.29	1.23	1.52	0.38	1.38	1.76	0.49	1.57	2.06

TABLE 3

No.	Items	Operating cost in cents per kw.-hr. at full load								
		A = 20,000 kw.			B = 5,000 kw.			C = 1,000 kw.		
		Constant part	Variable part	Total	Constant part	Variable part	Total	Constant part	Variable part	Total
27	Coal (19–25)	0.047	0.199	0.246	0.062	0.223	0.285	0.079	0.255	0.334
28	Water (13)	0.001	......	0.001	0.001	......	0.001	0.002	......	0.002
29	Wages	0.008	......	0.008	0.011	......	0.011	0.016	......	0.016
30	Repairs	0.004	......	0.004	0.005	......	0.005	0.008	......	0.008
31	Interest and renewals	0.049	......	0.049	0.065	......	0.065	0.098	......	0.098
	Total	0.109	0.199	0.308	0.144	0.223	0.367	0.203	0.255	0.458

TABLE 4

Items	Units	Symbol	Power plant A 20,000 kw.	Power plant B 5,000 kw.	Power plant C 1,000 kw.
Heat consumption at no load (25)	B.t.u. per kw.-hr.	a_w	3,900	5,150	6,580
Additional heat consumption (25)	B.t.u. per kw.-hr.	b_w	16,600	18,600	21,220
Operating cost without coal (32) (27)	Cents per kw.-hr.	c	0.062	0.082	0.124
Cost of coal (19)	Cents per mill B.t.u.	g	12	12	12

TABLE 5

No.	Items Units	Power plant A = 20,000 kw.	Power plant B = 5,000 kw.	Power plant C = 1,000 kw.
1	Momentary heat consumption, B.t.u./kw.-hr.	$W_t = 3{,}900\frac{1}{m} + 16{,}600$	$W_t = 5{,}150\frac{1}{m} + 18{,}600$	$W_t = 6{,}580\frac{1}{m} + 21{,}220$
4	Average yearly heat consumption, B.t.u./kw.-hr.	$W_m = 3{,}900\frac{1}{n}f + 16{,}600$	$W_m = 5{,}150\frac{1}{n}f + 18{,}600$	$W_m = 6{,}580\frac{1}{n}f + 21{,}220$
5	If $(f = 1)$, B.t.u./kw.-hr.	$W_{m1} = 3{,}900\frac{1}{n} + 16{,}600$	$W_{m1} = 5{,}150\frac{1}{n} + 18{,}600$	$W_{m1} = 6{,}580\frac{1}{n} + 21{,}220$
6	If $(f = n)$, B.t.u./kw.-hr.	$W_{m2} = 20{,}500$	$W_{m2} = 23{,}750$	$W_{m2} = 27{,}800$
7	Average yearly operating cost, cents/kw.-hr.	$K = 0.062\frac{1}{n} + \frac{12 \times W_m}{1{,}000{,}000}$	$K = 0.082\frac{1}{n} + \frac{12 \times W_m}{1{,}000{,}000}$	$K = 0.124\frac{1}{n} + \frac{12 \times W_m}{1{,}000{,}000}$

TABLE 6

Use factor	Limit $f = 1$, $n = m$ Equation #5, B.t.u. per kw.-hr.			Percentage 20,500 = 100 %			Limit $f = n$ Equation #6, B.t.u. per kw.-hr.		
	A	B	C	A	B	C	A	B	C
1.0	20,500	23,750	27,800	100.0	86.1	74.0	20,500	23,750	27,800
0.9	20,930	24,320	28,520	98.0	84.2	72.1	20,500	23,750	27,800
0.8	21,470	25,020	29,420	95.5	81.9	69.6	20,500	23,750	27,800
0.7	22,170	24,950	30,620	91.8	79.0	67.0	20,500	23,750	27,800
0.6	23,100	27,170	32,170	88.9	75.6	63.8	20,500	23,750	27,800
0.5	24,400	28,900	34,420	84.1	71.0	59.8	20,500	23,750	27,800
0.4	26,350	31,500	37,670	77.9	65.2	54.7	20,500	23,750	27,800
0.3	29,600	35,800	43,120	69.4	57.2	47.5	20,500	23,750	27,800
0.2	36,100	44,400	54,020	56.7	46.4	37.9	20,500	23,750	27,800
0.1	55,600	70,100	86,920	36.8	29.3	23.6	20,500	23,750	27,800

TABLE 7

A $\frac{0.062}{n}$	B $\frac{0.082}{n}$	C $\frac{0.124}{n}$	A $\frac{12 \times W_m}{1,000,000}$	B $\frac{12 \times W_m}{1,000,000}$	C $\frac{12 \times W_m}{1,000,000}$
0.062	0.082	0.124	0.246	0.285	0.334
0.069	0.091	0.138	0.252	0.292	0.342
0.078	0.102	0.155	0.258	0.300	0.352
0.089	0.117	0.177	0.266	0.312	0.368
0.103	0.137	0.207	0.278	0.326	0.386
0.124	0.164	0.248	0.293	0.347	0.413
0.155	0.205	0.310	0.316	0.378	0.452
0.206	0.273	0.413	0.356	0.430	0.517
0.310	0.410	0.620	0.433	0.535	0.650
0.620	0.820	1.240	0.667	0.840	1.040

TABLE 7 (*Continued*)

Use factor	Limit $f = 1$ Equations 5 and 7			Limit $f = n$ Equations 6 and 7		
	Power plant			Power plant		
	A 20,000 kw. cts./kw.-hr.	B 5,000 kw. cts./kw.-hr.	C 1,000 kw. cts./kw.-hr.	A 20,000 kw. cts./kw.-hr.	B 5,000 kw. cts./kw.-hr.	C 1,000 kw. cts./kw.-hr.
1.0	0.308	0.370	0.458	0.308	0.369	0.458
0.9	0.321	0.385	0.480	0.315	0.379	0.472
0.8	0.336	0.406	0.507	0.324	0.390	0.489
0.7	0.357	0.431	0.545	0.335	0.408	0.511
0.6	0.381	0.463	0.593	0.349	0.425	0.541
0.5	0.417	0.514	0.661	0.370	0.452	0.582
0.4	0.471	0.586	0.762	0.401	0.493	0.644
0.3	0.562	0.706	0.930	0.452	0.564	0.747
0.2	0.743	0.948	1.270	0.556	0.703	0.954
0.1	1.287	1.660	2.280	0.866	1.105	1.574

Annual Cost of Power.—Harrington Emerson says:[1]

"In China men are paid $0.01 an hour for climbing treadmills actuating stern wheels which propel river boats. These Coolies convert their stored human muscular energy into mechanical foot-pounds. From experience with treadmills in British prisons we know exactly the mechanical equivalent of hard labor. It is a climb of 8,640 ft. each 24 hr. This is the limit of human endurance for a succession of days. To convert this into horsepower we must know the man's weight and the number of hours he works each day. The average weight of man is about 150 lb. A man of this weight climbing 8,640 ft. in 24 hr. yields 1,296,000 ft.-lb. A horsepower for 24 hr. is 47,530,000 ft.-lb. It would therefore take 36.6 Chinamen to yield a continuous horsepower and the wages of these Chinamen would amount to $3.66 per day, or $1,336 a year.

"From Niagara you can buy a horsepower year for $20. It costs the paper mills which have their own power about $12 a year for continuous horsepower.

[1] *Proceedings* S.P.E.E., vol. 20, 1912.

Human energy at $0.01 a day costs one hundred and ten times as much as this water-power energy, although the supervising human labor receives an average of $3 a day.

"The substitution of uncarnate energy for human muscular energy has increased wages thirty-fold and has cheapened power to 1 per cent. of its cheapest muscular price. This is not all. When men are used as power generators the supply is strictly limited and can be easily monopolized. Uncarnate energy is

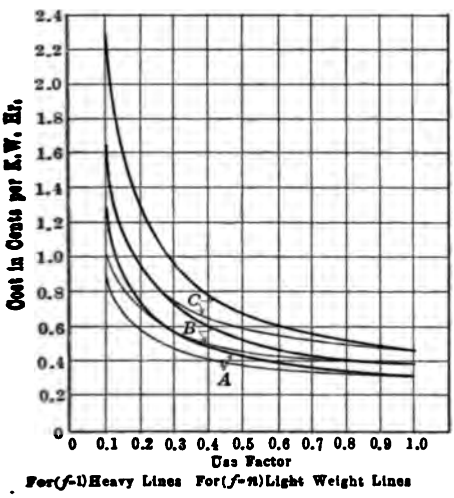


FIG. 199.—Klingenberg's cost curves.

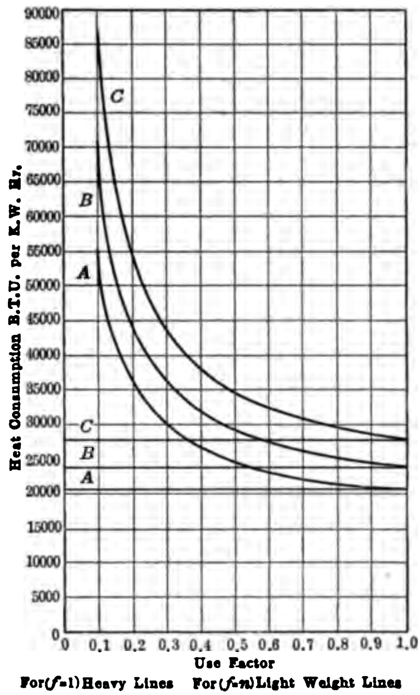


FIG. 200.—Klingenberg's heat consumption curve.

without limit as long as there is coal and oil and gas, as long as the sun shines and makes organic fuels or draws up water from the surface of the ocean.

"For strictly limited horsepower at $1,336 a year we now have unlimited horsepower at a minimum price of $12."

Although the cost of a horsepower varies greatly with local conditions and with the cost of fuel, water, labor and supplies yet a fair idea of the average cost may be obtained from the following tables prepared by different students of power costs.

Yearly Costs of Steam Power, 308 Days, 10 Hr. per Day, Simple Non-condensing Engine

Table A.—Engine and Boiler Combined

1. Horsepower of engine	2	3	4	6	8	10	12
2. Total coal consumption in pounds per horsepower-hour	13.0	10.50	8.50	7.90	7.60	7.40	7.25
3. Cost of plant per horsepower	200.00	152.00	133.00	110.00	89.00	83.00	78.00
4. Fixed charges on plant at 11 per cent $	44.00	50.00	58.20	72.50	78.20	91.50	102.00
5. Cost of coal at $5 per long ton $	180.00	215.00	233.00	325.00	420.00	510.00	600.00
6. Attendance $	99.00	109.00	116.00	136.00	154.00	173.00	184.00
7. Oil, waste and supplies $	13.20	14.30	15.40	17.60	20.00	22.00	23.00
8. Total yearly cost, coal at $5 per ton $	336.00	388.00	424.00	550.00	672.00	796.00	910.00
9. Total yearly cost, coal at $4 per ton $	300.00	345.00	385.00	495.00	610.00	720.00	810.00
10. Total yearly cost, coal at $3 per ton $	265.00	300.00	340.00	430.00	530.00	630.00	710.00
11. Yearly cost per horsepower, coal at $5 per ton $	168.00	130.00	106.00	92.00	84.00	79.60	76.00
12. Yearly cost per horsepower, coal at $4 per ton $	152.00	116.00	95.00	81.00	76.00	72.00	68.00
13. Yearly cost per horsepower, coal at $3 per ton $	132.00	102.00	83.00	72.00	66.00	63.00	59.00

Table B.—Engine and Boiler—Independent

1. Horsepower of engine	10	12	14	15	20
2. Total coal consumption in pounds per horsepower-hour	7.40	7.25	7.00	6.50	6.00
3. Cost of plant per horsepower $	10.00	194.00	182.00	174.00	158.00
4. Fixed charges on plant at 11 per cent $	230.00	255.00	280.00	285.00	337.00
5. Cost of coal at $5 per long ton $	510.00	600.00	675.00	690.00	830.00
6. Attendance $	173.00	184.00	194.00	202.00	230.00
7. Oil, waste and supplies $	22.00	23.80	25.80	26.50	31.20
8. Total yearly cost, coal at $5 per ton $	935.00	1,063.00	1,175.00	1,203.00	1,428.00
9. Total yearly cost, coal at $4 per ton $	840.00	960.00	1,050.00	1,080.00	1,260.00
10. Total yearly cost, coal at $3 per ton $	740.00	830.00	920.00	950.00	1,100.00
11. Yearly cost per horsepower, coal at $5 per ton $	93.50	88.00	83.00	80.00	71.00
12. Yearly cost per horsepower, coal at $4 per ton $	84.00	79.00	74.00	72.00	64.00
13. Yearly cost per horsepower, coal at $3 per ton $	74.00	68.00	64.00	62.00	56.00

Table B (*Continued*)

1. Horsepower of engine	30	40	50	75
2. Total coal consumption in pounds per horsepower-hour	5.50	4.75	4.50	4.00
3. Cost of plant per horsepower $	126.00	107.00	96.00	79.00
4. Fixed charges on plant at 11 per cent $	415.00	475.00	525.00	650.00
5. Cost of coal at $5 per long ton $	1,100.00	1,310.00	1,540.00	2,050.00
6. Attendance $	287.00	338.00	390.00	520.00
7. Oil, waste and supplies $	41.50	51.00	61.50	86.00
8. Total yearly cost, coal at $5 per ton $	18.43	2,194.00	2,516.00	3,306.00
9. Total yearly cost, coal at $4 per ton $	1,660.00	1,960.00	2,250.00	3,000.00
10. Total yearly cost, coal at $3 per ton $	1,450.00	1,710 00	1,960.00	2,650.00
11. Yearly cost per horsepower, coal at $5 per ton $	60.00	55.00	50.00	44.00
12. Yearly cost per horsepower, coal at $4 per ton $	54.00	49.00	45.00	39.00
13. Yearly cost per horsepower, coal at $3 per ton $	47.00	42.00	39.00	34.00

TABLE C

1. Horsepower of engine	10	12	14	15	20
2. Total coal per horsepower per hour, pounds	7.00	6.75	8.50	6.00	5.50
3. Cost of plant per horsepower $	220.00	204.00	192.00	186.00	163.00
4. Fixed charges on plant at 11 per cent $	242.00	270.00	295.00	307.00	360.00
5. Cost of coal at $5 per long ton $	480.00	560.00	625.00	670.00	750.00
6. Attendance $	178.00	190.00	202.00	210.00	238.00
7. Oil, waste and supplies $	22.80	24.80	26.70	27.60	32.50
8. Total yearly cost, coal at $5 per ton $	923.00	1,045.00	1,149.00	1,215.00	1,380.00
9. Total yearly cost, coal at $4 per ton $	830.00	940.00	1,030.00	1,100.00	1,240.00
10. Total yearly cost, coal at $3 per ton $	730.00	820.00	900.00	960.00	1,080.00
11. Yearly cost per horsepower, coal at $5 per ton $	92.30	87.00	82.00	80.00	69.00
12. Yearly cost per horsepower, coal at $4 per ton $	83.00	78.00	74.00	72.00	62.00
13. Yearly cost per horsepower, coal at $3 per ton $	73.00	68.00	65.00	63.00	54.00

TABLE C (*Continued*)

1. Horsepower of engine	30	40	50	75	100
2. Total coal per horsepower per hour, pounds	5.25	4.75	4.25	3.70	3.50
3. Cost of plant per horsepower $	134.00	120.00	108.00	93.00	81.00
4. Fixed charges on plant at 11 per cent $	440.00	530.00	590.00	765.00	890.00
5. Cost of coal at $5 per long ton $	1,040.00	1,310.00	1,470.00	1,910.00	2,420.00
6. Attendance $	297.00	350.00	405.00	535.00	760.00
7. Oil, waste and supplies $	43.00	53.00	64.00	89.00	114.00
8. Total yearly cost, coal at $5 per ton $	1,720.00	2,243.00	2,529.00	3,299.00	4,094.00
9. Total yearly cost, coal at $4 per ton $	1,550.00	2,020.00	2,270.00	2,961.00	3,700.00
10. Total yearly cost, coal at $3 per ton $	1,360.00	1,770.00	2,010.00	2,600.00	3,250.00
11. Yearly cost per horsepower, coal at $5 per ton $	57.00	56.00	51.00	44.00	41.00
12. Yearly cost per horsepower, coal at $4 per ton $	51.00	50.00	46.00	39.40	37.00
13. Yearly cost per horsepower, coal at $3 per ton $	44.50	43.50	40.00	34.50	32.50

YEARLY COST OF STEAM POWER, 308 DAYS, 10 HR. PER DAY, COMPOUND CONDENSING ENGINE

TABLE D

1. Horsepower of engine	100	200	300	400
2. Total coal per horsepower-hour, pounds	2.75	2.45	2.40	2.35
3. Cost of plant per horsepower $	105.00	93.30	86.40	76.20
4. Fixed charges on plant at 11 per cent $	1,160.00	2,060.00	2,850.00	3,350.00
5. Cost of coal at $5 per long ton $	1,910.00	3,370.00	5,100.00	6,700.00
6. Attendance $	880.00	1,220.00	1,220.00	1,760.00
7. Oil, waste and supplies $	143.00	205.00	240.00	285.00
8. Total yearly cost, coal at $5 per ton $	4,198.00	6,948.00	9,496.00	12,171.00
9. Total yearly cost, coal at $4 per ton $	3,780.00	6,200.00	8,550.00	11,000.00
10. Total yearly cost, coal at $3 per ton $	3,300.00	5,400.00	7,500.00	9,700.00
11. Yearly cost per horsepower, coal at $5 per ton $	42.20	35.10	31.50	30.50
12. Yearly cost per horsepower, coal at $4 per ton $	37.80	31.50	28.40	27.00
13. Yearly cost per horsepower, coal at $3 per ton $	33.20	27.70	25.00	23.80

TABLE D (*Continued*)

1. Horsepower of engine		500	600	700	800
2. Total coal per horsepower-hour, pounds		2.30	2.25	2.20	2.16
3. Cost of plant per horsepower	$	71.20	67.30	64.40	62.20
4. Fixed charges on plant at 11 per cent	$	3,920.00	4,451.00	4,952.00	5,492.00
5. Cost of coal at $5 per long ton	$	8,380.00	9,650.00	11,000.00	12,500.00
6. Attendance	$	1,930.00	2,100.00	2,650.00	2,700.00
7. Oil, waste and supplies	$	315.00	350.00	385.00	420.00
8. Total yearly cost, coal at $5 per ton	$	14,596.00	16,818.00	19,059.00	21,674.00
9. Total yearly cost, coal at $4 per ton	$	13,200.00	15,700.00	17,200.00	19,500.00
10. Total yearly cost, coal at $3 per ton	$	11,500.00	13,200.00	15,200.00	17,100.00
11. Yearly cost per horsepower, coal at $5 per ton	$	29.20	27.70	27.30	26.10
12. Yearly cost per horsepower, coal at $4 per ton	$	26.10	24.90	24.60	23.50
13. Yearly cost per horsepower, coal at $3 per ton	$	23.00	21.90	21.50	20.60

TABLE D (*Continued*)

1. Horsepower of engine		900	1,000	1,500	2,000
2. Total coal per horsepower-hour, pounds		2.10	2.00	1.80	1.75
3. Cost of plant per horsepower	$	59.30	55.70	54.40	53.20
4. Fixed charges on plant at 11 per cent	$	5,910.00	6,130.00	9,000.00	11,880.00
5. Cost of coal at $5 per long ton	$	14,300.00	14,500.00	18,600.00	24,200.00
6. Attendance	$	2,930.00	3,480.00	4,400.00	5,200.00
7. Oil, waste and supplies	$	445.00	470.00	600.00	685.00
8. Total yearly cost, coal at $5 per ton	$	23,644.00	24,595.00	35,100.00	42,018.00
9. Total yearly cost, coal at $4 per ton	$	21,200.00	22,200.00	31,500.00	37,800.00
10. Total yearly cost, coal at $3 per ton	$	18,500.00	19,500.00	27,500.00	33,000.00
11. Yearly cost per horsepower, coal at $5 per ton	$	25.20	24.50	23.50	21.00
12. Yearly cost per horsepower, coal at $4 per ton	$	22.60	22.00	20.30	18.90
13. Yearly cost per horsepower, coal at $3 per ton	$	19.90	19.40	17.90	16.60

Another estimate is presented in Table "E." This table was prepared by Mr. Webber in 1903.

COST OF STEAM POWER PER INDICATED HORSEPOWER OF 3,080 HR.

TABLE E

1. Horsepower of plant	100	200	300	400	500	600
2. Total plant per i.hp. including buildings	$170.00	$146.00	$126.00	$110.00	$96.00	$85.00
3. Fixed charges, 14 per cent	23.80	24.40	17.65	15.40	13.45	11.90
4. Coal per hp.-hr., lb.	7.00	6.00	6.00	5.50	5.00	4.50
5. Cost of coal at $4	38.50	35.70	33.00	32.00	27.50	24.70
6. Attendance	12.00	10.00	8.60	7.25	6.20	5.40
7. Oil, waste and supplies	2.40	2.00	1.72	1.45	1.24	1.08
8. Total with $4 coal	76.70	68.10	60.97	56.10	48.39	43.08
9. Total with $2 coal	57.45	50.25	44.47	40.10	34.64	30.73

TABLE E (*Continued*)

1. Horsepower of plant	700	800	900	1,000	1,500	2,000
2. Total plant per i.hp. including buildings	$76.00	$69.00	$64.00	$60.00	$58.00	$56.00
3. Fixed charges, 14 per cent	10.65	9.65	8.95	8.40	8.12	7.85
4. Coal per hp.-hr., lb.	4.00	3.50	3.00	2.50	2.00	1.50
5. Cost of coal at $4	22.00	19.20	16.50	13.75	11.00	8.25
6. Attendance	4.70	4.15	3.75	3.50	3.25	3.00
7. Oil, waste and supplies	0.94	0.83	0.75	0.70	0.65	0.60
8. Total with $4 coal	38.29	33.83	29.95	26.35	23.02	19.75
9. Total with $2 coal	27.29	24.23	21.75	19.47	17.52	15.50

$60 per horsepower-year at the machine is a common assumption where steam is the motive power.

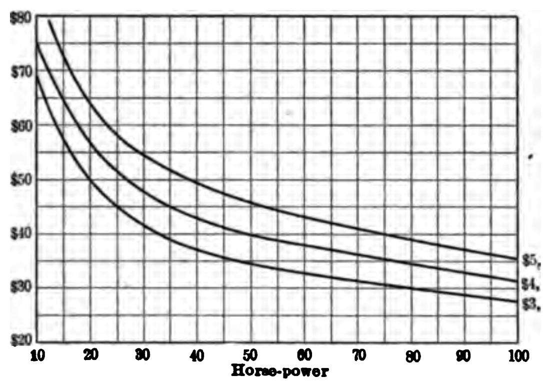


FIG. 201.—Cost of producing a horsepower-year of 3080 hours with simple non-condensing stationary engines with coal at $3.00, $4.00 and $5.00 per long ton.

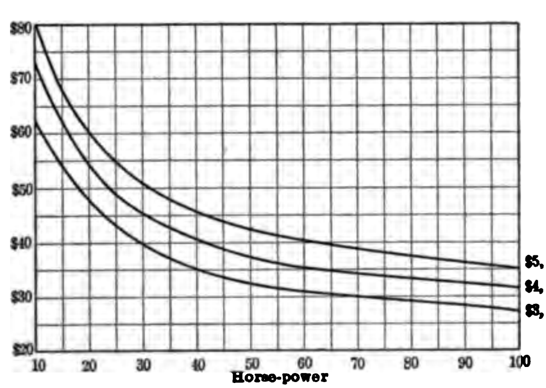


FIG. 202.—Cost of producing a horsepower-year of 3080 hours with simple condensing engines with coal at $3.00, $4.00 and $5.00 per long ton.

The accompanying diagrams show other estimates of the yearly cost of producing steam power under various conditions and costs of coal when running 10 hr. a day for 6 days a week with fairly steady load. They are intended to show the expense of running under everyday conditions on such a plant as a prudent man would install with ordinary skill.

Cost of 24-hr. power for 365 days per year is about 2.2 times the cost for 10-hr. power for 308 days.

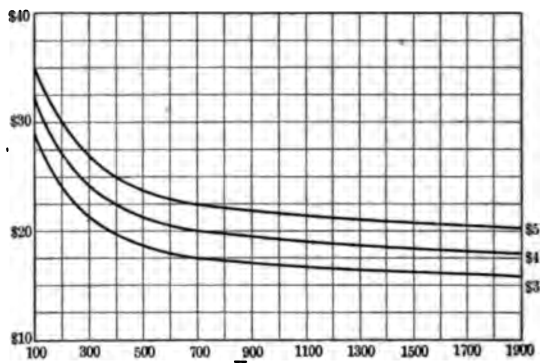


Fig. 203.—Cost of producing a horsepower-year of 3080 hours with compound condensing engines with coal at \$3.00, \$4.00 and \$5.00 per long ton.

Cost of 24-hr. power for variable load cannot be stated without knowing all conditions. For varying load, usually add about 20 per cent. to coal consumption required for steady load.

CHAPTER XV

HINTS ON STEAM PLANT OPERATION

The following hints regarding the operation of steam power plants are given with the idea that they may be of service to the young engineer during the trials and tribulations that come with early responsibility.

The operation or running of steam power plants with the accumulated experience of more than a century and a half might seem to most any one who has not been in the actual working to be a very simple problem, which throughout the last century at least had been standardized and was well understood. Indeed by talking to many operators whose duty it is to run steam plants, he would be sure that the last word had been said and that there was nothing to learn from further study. When one commences to investigate these plants from which such rose-colored reports are received and to note down and compare the figures, he will find discrepancies of a very serious nature, loop holes that are very wide indeed and he will soon find that his information is largely a matter of guesswork. If the investigator is an engineer, his scientific training will show him that the reported results are impossible and in searching around for the reasons he will find that his informant either is ignorant of the proper methods to apply and thus fooling himself with incorrect results, or he is simply hiding his own lack of knowledge by giving results he knows are not true.

The man who reports a horsepower-hour on 1 lb. of coal or an evaporation of 14 lb. of water per pound of coal, is in the class with the man who sprinkles a small amount of some chemical on ashes and reports that he gets more heat from the ashes than from an equal amount of good coal. This same man is likely to tell you his furnace temperature is 3,200° and his flue gases 200°.

The object of a power plant is to furnish power. When a power plant fulfills its purposes it must furnish power when it is wanted and in such quantities as wanted. It must be so run that power must be available whenever it is needed, that is, continuity of operation is essential and finally it must make and deliver power at as low a cost as is consistent with the circumstances of its design and location.

To fulfill the above conditions the operator must study the plant as a whole and in detail with reference to the following:

1. The plant must be studied to discover its strong points and weaknesses, that is, how well the designer has attacked the problem. What

has he forgotten or left out and how can the omissions be corrected? What particular parts cost more than the return from them warrants?

2. Each piece of machinery must be gone over and a proper understanding obtained of what may be expected of it. Is it fitted for the work it has to do? What would it cost to replace it with the best machine of its kind? Does it pay to run it? How should it be run to get the best results?

3. Fuel, being the largest single item in station cost, should receive the most careful study. A study of the fuel supply of the district is most profitable and may show that you can obtain a better and cheaper supply of fuel without difficulty.

Look into the methods of shipping and marketing the coal. It may be possible to buy the same coal in a different way to better advantage. How much water does the coal contain when delivered and what do you pay for the water? Look into the methods of storing, does the water evaporate or increase in the storage? Every pound of water evaporated from the coal in the furnace costs nearly as much to evaporate as if it were in the boiler and it is then wasted. Are the coal-handling arrangements bad or good? Sometimes conveyors cost more than coal handling with a horse and cart, or even a few sections of industrial railway.

The percentage of refuse is important as well as the combustible in the ash. If the refuse is more than a few per cent. more than the ash content, something is wrong with the grates or with the firing. If the combustible in the ash is high, over 50 per cent., the firing is bad or the cleaning of the fires has not been done properly. Be careful that your samples are average samples. Many incorrect results are due to imperfect sampling. Watch the combustion and air supply. Carbonic-acid determinations from beyond the bridge-wall and in the flue should be regularly made. The apparatus is simple and cheap and when carefully made the results are valuable. The results will not be absolute, however, but comparable. For accurate results, very careful sampling and analysis are required and these are not usually obtainable except in plants of the maximum size where one man is trained to do only this work. It is quite difficult for anyone to duplicate results.

4. *Feed Water.*—Is it pure or a good boiler water? A good boiler water is not necessarily pure. In fact pure water is too good a solvent and is not desirable. Is boiler compound needed? Do you buy patent medicine? The usual patented compounds are 50 cts. worth of lime and soda ash dissolved in a barrel of water and sold for 30 to 60 cts. per gallon. Learn what the scaling salts are and how they occur in the boiler water; and use the least amount of soda ash and lime that will soften the water. A little scale is not a bad thing. How much water do you blow off in the blowoff system? How much from the safety valves. What is the

leakage loss in the pipe system. Is there another source of feed water and how much does it cost? Are your water meters correct? Do they run fast on light loads? A known plant paid 40 per cent. more for water than it should, due to incorrect meters, so check them against weighing devices or a meter of the Venturi type once in a while. Are the steam pipes covered? What do you lose by condensation in the pipes? Are your drips thrown away or do they leak away? All clean drips should go to the boilers and there should be few dirty drips.

5. *Oil.*—How do you lubricate the machinery in the plant? If you have an oil system, is it working well? How much oil do you use? How is it purchased? How do you check up the quality of the oil? Some operators buy on a kilowatt-hour basis. Other operators buy by the gallon or barrel. Do you use compounded oil? You will hear some operators claim that cylinder oil will not lubricate unless it contains a certain proportion of tallow—say 10 or 12 per cent. This is fallacious but it dies hard. Do you oil through the carrying action of the incoming steam or do you feed the oil drop by drop *where* it is needed and *when* it is needed? How much care is exercised in saving oil?

Waste and Supplies.—How much waste do you use and how do you handle it? Will washable towels be better and cheaper than waste? Can you save oil that way? How do you look after packing gaskets and pump valves? Pump valves make good rubber heels. Do any of yours go that way? What small supplies do you keep and how are they issued and accounted for?

6. *Maintenance.*—In this particular item the operating man may save or lose quite a little money. Boilers must be cleaned, engines must be overhauled, pumps must be packed and valves replaced. Condensers must be cleaned and tubes replaced. If oil is present in the condensate the steam side of condensers should be boiled out with a soda solution occasionally. The dust must be blown out of electrical machinery. Dust and dirt must be kept out of everything and heat should be saved everywhere. Regular schedules of overhauling help, and if the work can be done between times, with the regular force, a saving will be made. How do you handle extraordinary repairs?

7. *Labor.*—Is your organization good or are you suffering from deadwood, dry rot, soldiering or incompetency. How do you run your force—on the team work principle or individual plan? Are your men ambitious for more work and responsibility or for more pay? Are they real mechanics or just workmen? Do they take pride in their work?

Finally.—Do you know the efficiency of your plant? Are you running on 8 lb. of coal per kilowatt-hour, or on, say, 2.5 lb. or better yet, do you turn out a kilowatt-hour for 35,000 B.t.u. in the fuel or 50,000 B.t.u., or even higher? Do you know your station water rate? Is it

20 lb. or 40 lb., and how accurately do you know it? What is your average boiler efficiency over a year, coal against water? What is your banking loss? Do you know your load factor and use factor? Can you by changing the use of your units improve your station economy? Which engine uses the most steam and which the least? Are the economy curves flat or deeply hollow? Do you know your costs? Who keeps them and are they kept accurately? Do you know what each piece of apparatus costs per year to keep it in good working operation? Costs are illusive and the operating man should look after them very carefully. How are your gages, thermometers and meters? Where and how were they rated? Do you keep standards to use for comparison? Are your valves tight? Do you keep the stems packed and what leakage do you have? Do you ever weigh your coal? How much does it lose in weight in the bunker? Do you wet down your coal?

CHAPTER XVI

POWER TRANSMISSION

Shafting and Belting.—The oldest and most common method of transmitting power from the engine to the consumer is by means of shafting, gearing and belting. Until within 25 years hardly any other method was considered except in special cases.

Chief Objection to this System.—The two chief objections to this system are its friction losses and its lack of adaptability. Experiments made by various engineers have shown losses between engine and machine in ordinary machine shops of from 50 to 60 per cent. of the total power transmitted by the engine. This loss is greatest in shops where large machines are employed, located at some distance from each other and it is here that other kinds of transmission can be used to advantage. Two incidental objections to shafting and belting are dirt and interference with proper lighting. These are objections of considerable importance in some shops as dirt and dust from the processes of manufacture are kept stirred up by the moving belts, often causing inconvenience to the workmen, and dark shops are not only objectionable from general standpoints but may lead to accidents.

Cost of Shafting and Belting.—The cost of shafting, including the necessary hangers, couplings and pulleys will vary according to size between \$2 and \$6 per linear foot. A rough rule which may be used in preliminary work is to allow \$1 per linear foot per inch of diameter.

The cost of belts will vary from \$5 to \$50 for such widths and lengths as are used in ordinary shop practice. The discount on all kinds of belting usually ranges from 50 to 70 per cent. of the list price.

Rope Driving.—Ropes may take the place of belts in special cases, but cannot be said, in any sense, to have replaced them for ordinary use. Manila and cotton ropes are sometimes used for main drives to connect the engine with the head shaft, and less frequently for distributing the power to the different floors.

It may be said in general that the first cost of ropes is less than of belts, but that they wear out much faster, are more difficult to splice, and are less efficient. They are sometimes valuable for carrying power at different angles.

Wire ropes have been used to a considerable extent in the past for carrying power comparatively long distances, especially in places exposed to the weather, but electricity has almost entirely supplanted them. The initial cost of a wire-rope transmission for distances from 300 to 1,000 ft. is very small, and the running expense is no greater than that of electricity but the rope drive is much more limited in its application.

Steam Transmission.—Where the area covered by an establishment is large it is often more economical to have a central boiler house and to transmit high-pressure steam to the various buildings, there to be used for both power and heating. The loss in transmission, although heavy, is probably much less than for shafting and belting if the pipes are of proper size and properly insulated.

Efficiency of Transmission.—Rather extravagant claims have been made as to the advantage of electricity over shafting and belting in the matter of efficiency. Experiments on several group installations in machine shops have shown a loss of from 40 to 60 per cent. of the total power of the engine before reaching the machine, as previously pointed out. These losses are due partly to the shafting and belting and would be reduced with independent motors. Direct tests on 16 large machines driven by independent motors in a locomotive works showed an average of 8.85 hp. for the machine and its work and 2.35 hp. for the power consumed by the motor and countershaft. This means an efficiency of less than 80 per cent. for the motors, not counting the losses in the generator and transmission lines. On the other hand, it may be said that although the friction losses in shafting and belting remain nearly constant at all loads, the electrical losses will diminish as the load falls off.

The following sections relating to "Electric Drive Versus Shafting and Belting" have been thoroughly revised and brought down to date by C. E. Clewell, Assistant Professor of Electrical Engineering at the University of Pennsylvania, whose articles in *American Machinist* (1914–15) relating to this subject are well known.

Electric Drive Versus Shafting and Belting.—The main advantages of the electric drive are included under the heads of "Location of Machines," "Head Room," "Centralized Power," "Reliability," and the "Ability to Study Machine Performance."[1]

Flexibility in the location of machinery with electric drive has come to be an acknowledged advantage in machine-shop work and particularly in the use of portable tools. The clear head room resulting from the use of individual motor-driven machines and the eliminating of overhead belting, adapts manufacturing spaces to improved lighting and ventilating conditions, and to more effective crane service, since the interference of overhead belts often dictates just what portions of a shop may be served by the crane and which may not. Furthermore, a centralized

[1] "American Handbook for Electrical Engineers," John Wiley and Sons, New York, p. 972.

ower station is made possible through the medium of electric power istribution, thus making changes and extensions of the plant practically ıdependent of the power supply. Under reliability, it follows that the reakdown of a single motor which is individually connected to its own ıachine tool, affects the operation of that machine only, whereas a breakown in the belting or shafting of a line-shaft drive, often causes interrupion for a larger group of machinery.

The study of machine performance which is a valuable accompaniıent to improved production methods, and to the application of so-called scientific management" to machinery practice, has practically been ıade possible for the first time, through the use of the recording and raphic electric meters which may be used in conjunction with indiidually motor-driven machinery. This feature of electric drive is beinning to be recognized as one of its most important advantages.

Methods of Motor Drive.—These may be classified as individual and s group drives. In the former, each machine is fitted with its own motor, ither driven directly or through a countershaft, and there is an entire bsence of overhead belts. This system is particularly applicable to hops having large machines located some distance apart and perhaps 'arying in character. It is necessary, however, in such cases to control he speed of the machine directly through the speed control of the motor, ınd this may be done by any one of the various methods of motor speed ontrol. Ranges of speeds of 4 to 1 are common, but in some special ases speed ranges as high as 10 to 1 are available for motor drive.

In the group method, several machines are arranged in a group and are driven by a short line shaft which is driven in turn by a motor. This makes it possible to use a constant-speed motor because the speed adjustments of the machine tool are effected in the ordinary manner through the medium of cone pulleys or gears. The size of the motor may also be smaller than that of the motors used for the corresponding machine tools with the method of individual motor drive, on account of the diversity actor of the group of machines.

On the other hand, with the group method of driving, the overhead belts are only partially done away with, and there is not that freedom of arrangement which makes the individual method of driving so desirable. It is also necessary in this connection to distinguish between the use of direct- and alternating-current motors. In general, individual motor drives necessitate direct-current motors, because of their adaptability o flexible speed control, while in the group method either direct- or lternating-current motors may be used since the motor under this conlition may usually be of the constant-speed type.

The choice of direct or alternating current for machine-shop drives lepends also to some extent upon whether there is a possibility or likelihood of throwing over, at certain times, to an outside power company's circuits. If the ready-to-serve charge is not too great, the alternating-current distribution from this one standpoint may be best, because the alternating current which is usually employed by the large central power stations, could thus be relied upon at certain times.

In the case of an individual shop power plant in which the amount of power required by the shop is large and there is no public service corporation to rely upon in the neighborhood, the question of direct or alternating current is partly dependent on the area covered by the plant, and partly dependent on the need for adjustable-speed motors. Alternating current is satisfactory both for small and for large plants from the viewpoint of distribution, but from this same viewpoint, direct current is hardly suitable for a plant extending over a considerable area, because of the large drop in voltage due to the long lines required and the relatively low supply voltages usually employed with direct-current systems of distribution. Adjustable-speed motors, however, are mainly of the direct-current type, and where their use is essential, it may be found desirable to employ alternating current for the main distribution circuits, and to transform from alternating to direct current by rotary converters or motor-generator sets, to meet the need of a source of direct current for motors of this type. In some plants, therefore, circuits of both types will be found, those of the alternating-current type being depended upon for lighting, and constant-speed motors, and those of the direct-current type supplying the adjustable-speed motors and sometimes certain electric lamps which are operative only on direct-current circuits.

Some years ago a great deal of discussion took place about the possibility of adjustable-speed motors for the various forms of machine-tool drive. Motors of this type are now available and are widely used. Machine-tool builders, who in the past have found it necessary to design their tools so as to get speed changes mechanically, now in many cases find it desirable to adapt the design of their machines for operation by individual electric motors.

The amount of power drawn for any given tool varies usually over a wide range, and a motor should be put on an individual tool which can take care of the largest load that the tool is apt to require, although its rated capacity need not be determined by this maximum demand on account of the liberal overloads which modern motors can develop for short intervals without excessive heating. In general, if a number of tools are grouped together, a motor that is appreciably smaller than the sum of the individual horsepower capacities required on the different tools may be installed. This is illustrated by the fact that even in the case of the group drive, the actual power drawn from the generator may be less than the total rated capacity of the motors connected to the generator. It must be remembered, however, that the power required to drive a machine tool is very small in proportion to the total cost of production chargeable to that tool, and hence other advantages of the individual motor drive may entirely offset the small gain in reduced motor size when the group method is employed in contrast to the individual method of drive.

In the installation of all motors, because of low efficiency at low loads, care should be taken that motors not too large for the work are chosen. With alternating-current motors, the fractional loads are also accompanied by low power factor; hence for this additional reason it is better to operate an alternating-current motor at or near its full rated capacity rather than at a load much below normal for a large part of its operation.

To summarize the matter of group versus individual methods of drive, it may be stated that there is still much difference of opinion regarding their relative advantages, although these differences of opinion are not so marked as was the case 10 years ago. Both methods are quite widely used, and each individual case requires careful study before an intelligent decision can be reached as to the actual merits of each method. In many cases both methods of driving will be found in the same plant.

Three items[1] stand out as most important in this question: (*a*) the influence of the character of the load; (*b*) the influence of speed; and (*c*) the influence of relative cost. Under (*a*) there is a general acceptance of the conclusion that where machines are operated intermittently, the method of individual drive is to be preferred. With the group method, the total load is made up partly of friction and other mechanical losses which go on continuously, and partly by the demand of the machine tools themselves when working. If, therefore, the load factor of the machines is low, the friction losses form a larger percentage of the total power consumed, and the group system thus becomes less efficient than where the machine load factor is high.

Under (*b*) the wide range and fine gradations of speed with an individual adjustable-speed motor give it a decided advantage in the matter of speeds over the group or line-shaft drive. The modern interpole, adjustable-speed motor possesses excellent commutation characteristics both at heavy and at light loads for a wide range of speeds, thus overcoming one of the larger difficulties in earlier types. Under (*c*) it may be said that the choice between individual and group drives is essentially one of relative cost. While the first cost of the motor equipment for individual driving is greater than the cost of the motor equipment for group driving, the economic returns through increased production by the use of individual motors may actually offset the higher first cost in a relatively short

[1] "American Handbook for Electrical Engineers," John Wiley and Sons, New York, p. 973.

time interval. Moreover, what may be termed secondary advantages, such as a more open shop space, better illumination and ventilation, and improved crane service, all form additional advantages in favor of the individual drive, which, while difficult to evaluate into cost equivalents, are now recognized as being of distinct economic value to any plant.

Sizes of Motors Recommended to Drive Machine Tools.—The accompanying tables contain the sizes and speeds of motors usually employed with the average duty indicated for machine tools.

The average load factor for motors driving lathes is from 10 to 25 per cent. On some special machines, as driving-wheel and car-wheel lathes, the cuts are all heavy, which increases the average load factor to from 30 to 40 per cent.

For extension boring mills, 5-hp. motors are used to move the housings on from 10-ft. to 16-ft. mills, 7½-hp. for from 14-ft. to 20-ft. mills and 10-hp. for from 16-ft. to 24-ft. mills. The load factor of the driving motor on boring mills averages from 10 to 25 per cent.

The load factor of motor-driven drills is about 40 per cent., when the larger drills applicable thereto are used. If the smaller drills are used, the load factor averages 25 per cent. and lower.

For the average milling operations the load factor averages from 10 to 25 per cent. On slab-milling machines where large quantities of metal are renewed it will average from 30 to 40 per cent.

On planers the load factor averages between 15 and 20 per cent. The motor must be large enough to reverse the bed quickly, yet this peak load occurs for such short intervals that it does not increase the average load per cycle very much.

The work done on shapers is of a varying character. With light work the load factor will not exceed 15 to 20 per cent.; with heavy work, the load factor will be as high as 40 per cent.

The conditions encountered on slotters are similar to those on shapers.

In the following tables[1] the horsepower recommended is based on average practice; it may therefore be decreased for very light work and must often be increased for heavy work. For convenience, the class of motor is indicated by the symbols *A*, *B* and *C*, which have the following meanings: (*A*) refers to the adjustable-speed shunt-wound direct-current motor, used wherever a number of different speeds are essential. (*B*) refers to the constant-speed shunt-wound direct-current motor, where the speeds are obtained by a gear-box or cone-pulley arrangement or where one speed only is required. (*C*) refers to the squirrel-cage induction motor for use in alternating-current circuits and used or adapted to those cases where direct current is not available. A gear-box or cone-pulley arrangement must be used to obtain different speeds.

[1] Based on the practice of the Westinghouse Electric and Manufacturing Co.

TABLE I.—ENGINE LATHES

Motor *A*, *B* or *C*

Swing, inches	Horsepower	
	Average	Heavy
12	1½	2
14	¾ to 1	2 to 3
16	1 to 2	2 to 3
18	2 to 3	3 to 5
20 to 22	3	7½ to 10
24 to 27	5	7½ to 10
30	5 to 7½	7½ to 10
32 to 36	7½ to 10	10 to 15
38 to 42	10 to 15	15 to 20
48 to 54	15 to 20	20 to 25
60 to 84	20 to 25	25 to 30

Axle Lathes

	Horsepower
Single	5, 7½, 10
Double	10, 15, 20

Wheel Lathes

Size, inches	Horsepower	Tail stock motor,[1] horsepower
48	15 to 20	5
51 to 60	15 to 20	5
79 to 84	25 to 30	5
90	30 to 40	5 to 7½
100	40 to 50	5 to 7½

[1] Standard machine-tool traverse motor.

TABLE II.—BOLT AND NUT MACHINERY

Bolt Cutters, Motor *A*, *B* or *C*

	Size, inches	Horsepower
Single	1, 1¼, 1½	1 to 2
	1¾, 2	2 to 3
	2¼, 3½	3 to 5
	4, 6	5 to 7½
Double	1, 1½	2 to 3
	2, 2½	3 to 5
Triple	1, 1½, 2	3 to 7½

Bolt Pointers, Motor, *B* or *C*

1½, 2½	1 to 2

TABLE II.—BOLT AND NUT MACHINERY.—(*Continued*)

Nut Tappers, Motor, *A*, *B* or *C*

Four-spindle	1, 2	3
Six-spindle	2	3
Ten-spindle	2	5

Nut Facing, Motor, *B* or *C*

1, 2	2 to 3

Bolt Heading, Upsetting and Forging, Motor, *A*,[1] *B*[2] or *C*[3]

Size, inches	Horsepower
¾ to 1½	5 to 7½
1½ to 2	10 to 15
2½ to 3	20 to 25
4 to 6	30 to 40

[1] Speed variation is sometimes desired when different sizes of bolts are headed on the same machine.

[2] Compound-wound direct-current motor.

[3] Wound secondary or squirrel-cage motor with approximately 10 per cent. slip.

TABLE III.—BORING AND TURNING MILLS

Motor, *A*, *B* or *C*

Size	Horsepower, Average	Horsepower, Heavy
37 to 42 in.	5 to 7½	7½ to 10
50 in.	7½	7½ to 10
60 to 84 in.	7½ to 10	10 to 15
7 to 9 ft.	10 to 15	
10 to 12 ft.	10 to 15	30 to 40
14 to 16 ft.	15 to 20	
16 to 25 ft.	20 to 25	

Drilling and Boring Machines, Motor, *A*, *B* or *C*

	Horsepower
Sensitive drills up to ½ in.	¼ to ¾
Upright drills, 12 to 20 in.	1
Upright drills, 24 to 28 in.	2
Upright drills, 30 to 32 in.	3
Upright drills, 36 to 40 in.	5
Upright drills, 50 to 60 in.	5 to 7½

	Horsepower, Heavy	Horsepower, Average
Radial drills, 3-ft. arm	3	1 to 2
Radial drills, 4-ft. arm	5 to 7½	2 to 3
Radial drills, 5 to 6 and 7-ft. arm	5 to 7½	3 to 5
Radial drills, 8 to 9 and 10-ft. arm	7½ to 10	5 to 7½

Cylinder Boring Machines, Motor, *A*, *B* or *C*

Diameter of spindle, inches	Max. boring diam., inches	Horsepower
4	20	7½
6	30	10
8	40	15

TABLE III.—BORING AND TURNING MILLS.—(*Continued*)

Pipe Threading and Cutting-Off Machines, Motor, *A*, *B* or *C*

Size pipe, inches	Horsepower
1/4 to 2	2
1/2 to 3	3
1 to 4	3
1 1/4 to 6	3 to 5
2 to 8	3 to 5
3 to 10	5
4 to 12	5
8 to 18	7 1/2
24	10

TABLE IV.—BULLDOZERS OR FORMING OR BENDING MACHINES

Motor, B[1] or *C*[2]

Width, inches	Head movement, inches	Horsepower
29	14	5
34	16	7 1/2
39	16	10
45	18	15
63	20	20

Buffing Lathes, Motor, *B* or *C*

Wheels No.	Diam., inches	Horsepower
2	6	1/4 to 1/2
2	10	1 to 2
2	12	2 to 3
2	14	3 to 5

[1] Compound-wound motor.
[2] Wound secondary or squirrel-cage motor with approximately 10 per cent. slip.

For brass tubing and other special work use about double the above horsepower.

TABLE V.—PLANERS

Motor, *A*,[1] *B*[1] or *C*

Width, inches	Under rail, inches	Horsepower
22	22	3
24	24	3 to 5
27	27	3 to 5
30	30	5 to 7 1/2
36	36	10 to 15
42	42	15 to 20
48	48	15 to 20
54	54	20 to 25
60	60	20 to 25
72	72	25 to 30
84	84	30
100	100	40

Normal length of bed in feet is about one-fourth the width in inches.

[1] Compound-wound motor.

TABLE V.—PLANERS.—(*Continued*)

Rotary Planers, Motor, *A*, *B* or *C*

Diam. of cutter, inches	Horsepower
24	5
30	7½
36 to 42	10
48 to 54	15
60	20
72	25
84	30
96 to 100	40

TABLE VI.—HYDROSTATIC WHEEL PRESSES

Motor, *B* or *C*

Size, tons	Horsepower
100	5
200	7½
300	7½
400	10
600	15

TABLE VII.—PUNCHING AND SHEARING MACHINES

Presses for Notching Sheet Iron. Motor, *A*, *B* or *C*, ½ to 3 hp.

Punches. Motor, *B*[1] or *C*[2]

Diam., inches	Thickness, inches	Horsepower
⅜	¼	1
½	½	2 to 3
⅝	⅝	2 to 3
¾	¾	3 to 5
⅞	¾	5
1	½	5
1	1	7½
1¼	1	7½ to 10
1½	1	10 to 15
2	1	10 to 15
2½	1½	15 to 25

[1] Compound-wound motor.

[2] Wound secondary or squirrel-cage motor with approximately 10 per cent. on the larger sizes.

Shears. Motor, *B*[1] or *C*[2]

	Horsepower	
Width, inches	Cut, ¼ in. iron	Cut, ½ in. iron
30 to 42	3	5
50 to 60	4	7½
72 to 96	5	10

Bar shears . . . . 7½ hp.
Double-angle shears . . . . 10 hp.

[1] Compound-wound motor.

[2] Wound secondary or squirrel-cage induction motor with 10 per cent. slip.

TABLE VII.—PUNCHING AND SHEARING MACHINES.—(*Continued*)

Lever Shears, Motor, B^1 or C^2

Size, inches	Horsepower
1 × 1	5
1½ × 1½	7½
2 × 2	10
6 × 1	
2½ × 2½	15
1 × 7	
2¾ × 2¾	20
1½ × 8	
3½ × 3½	30
4½ round	

[1] Compound-wound motor.
[2] Wound second or squirrel-cage motor with approximately 10 per cent. slip.

Plate Shears, Motor, B^1 or C^2

Size of metal cut, inches	Cut per minute	Length of stroke, inches	Horsepower
⅜ × 24	35	3	10
1 × 24	20	3	15
2 × 14	15	4¼	30
1 × 42	20	4	20
1½ × 42	15	4½	60
1¼ × 42	18	6	75
1½ × 72	20	5½	10
1¼ × 100	10 to 12	7½	75

[1] Compound-wound motor.
[2] Wound secondary or squirrel-cage motor with approximately 10 per cent. slip.

Plate Squaring Shears, Motor, *B* or *C*

Size of plates, inches	Cuts per minute	Horsepower
54 × 54	30	7½
3⁄16 packs		
72 × 72	30	7½
3⁄16 packs		

TABLE VIII.—SHAPERS

Motor, *A*, *B* or *C*

Stroke, inches	Horsepower, single head
12 to 16	2
18	2 to 3
20 to 24	3 to 5
30	5 to 7½

Traverse Head Shaper

20	7½
24	10

TABLE VIII.—SHAPERS.—(*Continued*)

Rolls—Bending and Straightening, Motor, B^1 or C^2

Width, feet	Thickness, inches	Horsepower
4	3/8	5
6	5/16	5
6	7/16	7½
6	3/4	15
8	7/8	25
10	1⅛	35
10	1½	50
24	1	50

[1] Standard bending roll motor.
[2] Wound secondary induction motor.

Saws, Cold and Cut Off, Motor, *A*, *B* or *C*

Size of saw, inches	Horsepower	Size of saw, inches	Horsepower
20	3	36	10 to 15
26	5	42	20
32	7½	48	25

Slotting and Key Seating, Motor, *A*, *B* or *C*

Stroke, inches	Horsepower	Stroke, inches	Horsepower
6	3	16	7½
8	3 to 5	18	7½ to 10
10	5	20	10 to 15
12	5	24	10 to 15
14	5 to 7½	30	10 to 15

Horizontal Boring, Drilling and Milling Machines, Motor, *A*, *B* or *C*

Size of spindle, inches	Horsepower for single spindle
3½ to 4½	5 to 7½
4½ to 5½	7½ to 10
5½ to 6½	10 to 15

For machines with double spindles use motors of double the horsepower given.

TABLE IX.—MULTIPLE SPINDLE DRILL

Motor, *A*, *B* or *C*

Size of drills, inches	Max. No. drills to spindle	Horsepower
1/32 to 1/4	6	3
1/16 to 3/8	10	5
3/16 to 1/2	10	7½
1/4 to 3/4	10	10
3/8 to 1	10	10 to 15
2	4	7½
2	6	10
2	8	15

TABLE IX.—MULTIPLE SPINDLE DRILL.—(*Continued*)

Emery Wheels, Grinders, Etc., Motor, *B* or *C*

Wheels No.	Size, inches	Horsepower
2	6	½ to 1
2	10	2
2	12	3
2	18	5 to 7½
2	24	7½ to 10
2	26	7½ to 10

Miscellaneous Grinders, Motor, *B* or *C*

	Horsepower
Wet-tool grinder	2 to 3
Flexible swinging, grinding and polishing machine	3
Angle-cock grinder	3
Piston-rod grinder	3
Twist-drill grinder	2
Automatic-tool grinder	3 to 5

TABLE X.—MILLING MACHINES

Vertical Slabbing Machines, Motor, *A*, *B* or *C*

Width of work, inches	Horsepower
24	7½
32 to 36	10
42	15

Vertical Milling Machines

Height under work, inches	Horsepower
12	5
14	7½
18	10
20	15
24	20

Plain Milling Machines

Table feed, inches	Cross feed, inches	Vertical feed, inches	Horsepower
34	10	20	7½
42	12	20	10
50	12	21	15

Universal Milling Machines

Machine No.	Horsepower	Machine No.	Horsepower
1	1 to 2	3	5 to 7½
1½	1 to 2	4	7½ to 10
2	3 to 5	5	10 to 15

Horizontal Slab Millers

Width between housings, inches	Horsepower Average	Horsepower Heavy
24	7½ to 10	10 to 15
30	7½ to 10	10 to 15
36	10 to 15	20 to 25
60	25	50 to 60
72	25	75

TABLE XI.—GRINDING MACHINES (GRINDING SHAFTS, ETC.)

Motor, *A*, *B* or *C*

Diam. wheel, inches	Length work, inches	Horsepower Average work	Horsepower Heavy work
10	50	5	7½
10	72	5	7½
10	96	5	7½
10	120	5	7½
14	72	10	15
18	120	10	15
18	144	10	15
18	168	10	15

Gear Cutters, Motor, *A*, *B* or *C*

Size, inches	Horsepower	Size, inches	Horsepower
36 × 9	2 to 3	60 × 12	5 to 7½
48 × 10	3 to 5	72 × 14	7½ to 10
30 × 12	5 to 7½	64 × 20	10 to 15

Hammers, Motor, *B*[1] or *C*[2]

Size, pounds	Horsepower
15 to 75	½ to 5
100 to 200	5 to 7½

Bliss drop hammers require approximately 1 hp. for every 100-lb. weight of hammer head.

[1] Compound-wound motor.

[2] Wound secondary squirrel-cage motor with approximately 10 per cent. slip.

Selection of Motors and Speed Requirements for Machine Service.—The selection of the proper motor for given service requirements necessitates a careful study into the characteristics of the work to be performed by the machine tool. Both power and speed requirements vary widely in different tools, and a motor well adapted to one class of work may be unsuited to another. Expert advice should be secured in such a problem at least in the first cases until the factors involved are thoroughly understood, and the characteristics of the various types of motors on the market are mastered in their relation to the power and speed conditions to be supplied for given machines.

The importance of speed is at once realized when one considers that production depends to a great extent on the use of the most economical cutting speed for given operations. The 4 to 1 adjustable-speed motor now commonly employed in machine-shop work will be found to meet most conditions. It is recommended[1] that given machine tools be restricted to a certain range of diameters of work and thus make possible a close speed adjustment within this range, rather than to work a maximum range of diameters on a given tool with less refinements in the speed

[1] Westinghouse practice.

ıstment. Useful charts are available from the larger electrical manuıırers by means of which the horsepower requirements for given depths ııt, cutting speeds and feed, may be determined conveniently. Simiʀ, charts are available from which the relation between feeds, cutting ·ds, diameters of work, and spindle speeds may be determined conıently. The following table[1] is also useful in this general connection:

Metal	Horsepower per cubic foot of metal per minute
Cast iron	0.3 to 0.5
Wrought iron	0.6
Machinery steel	0.6
Steel, 50 carbon and harder	1.0 to 1.25
Brass and similar alloys	0.2 to 0.25

drills, the cubic inches of metal removed per minute are found by the nula:

$$Q = 0.7854 \times d^2f$$

·re d is the diameter of the drill in inches and f the feed in inches per ıute. The constants for drills are approximately double those given ·he previous table.

Cost of a Horsepower at the Machine.—The cost of power transted electrically to the consumer will depend on the original cost at engine or water power. Power transmitted from a large waterfall electricity frequently sells for $25 per horsepower-year for short disce transmission. (It is reported that Niagara power sells from $12 to per horsepower-year.)

Mr. W. C. Webber estimates the total cost at the machine of electrical ısmission in shops as follows, allowing for interest and depreciation including repairs, attendance, etc.:

Per horsepower-year of 3,080 hr.	$52
Per horsepower-year of 7,392 hr.	57
Per horsepower-year of 8,760 hr.	66

se are computed on a basis of 100 hp. produced at the engine for $35 horsepower-year at 3,080 hr.

Ɔne authority states that with coal at $3 per ton a simple non-consing, high-speed engine will produce 500 hp. at a cost of $36 per horseer-year of 3,080 hr., while a triple-expansion condensing engine will ıish the same power for $24.

Гhe probable costs in an electric system for furnishing power to a hine shop might be summarized as follows:

[1] Westinghouse Electric and Manufacturing Co. The table refers to lathes, ers, etc., when round-nose tools are used.

Cents per kilowatt-hour	
Coal at $3.50 per ton	0.5
Labor	0.4
General expenses	0.2
Fixed charge	0.4
Total	1.5

This does not allow for the cost or losses of distribution, but represents the cost at the switchboard, and corresponds very nearly to $35 per horse-power-year of 3,080 hr.

The following data has been compiled by the Bullock Electric Manufacturing Co. for the cost of power compared with output of the factory for different systems of transmission:

(a) Ordinary belting and shafting	1.7 to 2.0 per cent.
(b) Electric group drive	0.8 to 1.0 per cent.
(c) Individual motor drive	0.4 per cent.

It is evident that there is a marked saving in power through the use of individual motor drive, but it is also apparent that in most factories the important factor is increase in output, and if one system of power effects an increase of even 50 per cent. in the power consumption, this increase is relatively a small factor in the total cost of production. In this connection, it may be stated that there is abundant testimony that the installation of individual motor drive in large shops has effected from 20 to 40 per cent. increase in the output for given machine tools, and an equal or even greater economy in room.

Through the coöperation of Professor H. B. Dates the following notes relating to electrical machinery are given:

Direct Current versus Alternating Current.—For equal low voltages usually found in the small machine shop of say 250 volts, both systems require the same amount of copper in general terms in the matter of distribution about the buildings of the plant. If the three-wire system is used with the direct-current and the two-wire system with the alternating-current, the advantage is in favor of the direct-current system. The advantage of the alternating-current system lies in the flexible method of transformation from low to high voltages and *vice versa* by means of the transformer, thus permitting the use of high voltage for the power transmission at low current and consequently low line losses and low copper costs. Direct current is usually generated at comparatively low voltage and it cannot readily be transformed from one voltage to another as with the alternating current. Direct-current motors, however, are essential where adjustable-speed service is a requirement. Incandescent lamps may be operated on either direct- or alternating-current circuits. Lamps of the 110-volt class, however, are better than those designed for operation on 220-volt circuits.

gh voltages, which are desirable for economy in the transmission er over long distances, are more or less out of place in most manung plants, and consequently in the smaller plants where the disto be covered are relatively small, direct current is used. With voltage direct-current system, 110 or 220 volts, there are no transs to install and maintain, there is no danger from very high voltınd the system as a whole is characterized by simplicity both in and operation. As extensions are made to such a plant, however, fficulties in power transmission become greater due to the heavy losses at low voltage, and it may be desirable to use alternating t at a higher voltage. From these circuits, lamps and constantmotors may be supplied, and where adjustable-speed direct-current s must be used, special direct-current circuits may be employed, ansformation from alternating to direct current being effected by tary converter (also known as the synchronous converter) or by a -generator set.

rect-current Motors, Types and Where Used.—Shunt motors s good starting torque and favorable operating characteristics for operations. Their speed regulation is usually very good. They e used where the load at starting is fairly high and where practically nt speed is desired at all loads, *i.e.*, for line shafting, group drives, r many machine tools.

ies motors are used only where they can be geared or securely d to their load, to prevent the excessive speeds which result when ad on a series motor is thrown off. They are adapted for heavy ıg duty where variations in speed are not objectionable. In such s the speed varies approximately inversely as the square of the load. motors are adapted to street-car service, to cranes, hoists, fans and r loads.

mpound motors, cumulatively wound, have characteristics which d upon the relative strength of shunt and series field coils. These s have operating features which result from the combined effects ies and shunt windings, so that in a measure the action is partly ıat of a shunt motor and partly like that of a series motor. Cumu-wound compound motors possess a good starting torque, but are have poorer speed regulation, that is, a greater drop in speed with ses in load, than in the case of the straight shunt motor.

mpound-wound motors of the cumulative type are sometimes used ds where fairly large torque is required at one portion in each cycle here the load is relatively light for the rest of the cycle. They are quite extensively for elevator work, heavy planers, and for similar It should be noted that a portion of the torque in this motor is its series-motor characteristics, but when the speed increases, the field flux does not tend to decrease therewith as in the series motor, because the current in the shunt circuit remains practically constant. The employment of the shunt field in addition to a series field thus prevents the speed from exceeding safe limits at low loads.

Interpole or Commutating-pole Motors.—This type has come into very wide use during recent years for adjustable-speed machine-shop service. The interpoles render commutation practically perfect under wide ranges of speeds and loads. Adjustable-speed motors with wide speed ranges cost more than constant-speed motors of the same horsepower rating.

Temperature rise, rather than commutation, is the limiting factor in the output which can be supplied by a given motor. For constant-speed service the use of interpoles is not as essential as in the adjustable-speed service. In constant-speed service, however, where the load is subject to rapid and violent fluctuations and where very large overloads must be carried, the design should be such as to take care of the main heating effects, while interpoles may be employed to maintain perfect commutation under severe load conditions. Therefore, for intermittent constant-speed service, the interpole motor is more desirable than the motor of usual design.

In direct-current railway service the use of interpoles enables motors to withstand large loads for short periods, or, where artificial ventilation is used, to operate continuously at overloads without excessive sparking as a limiting factor. This statement also applies in a general way to series motors used for mill purposes.

Shunt motors, when used for adjustable-speed service, have their performance greatly improved with interpoles, and it is possible to reverse a 5-hp. shunt motor when interpoles are used, under full-load conditions, without sparking. This is not possible with ordinary shunt motors.

The interpole motor is simple in construction, requires but two service wires, works at maximum voltage, that is, from circuits of normal voltage, may have its speed control accomplished by the insertion of a rheostat in the field circuit in the ordinary manner, and operates throughout a large speed range at almost constant efficiency. Direct-current motors of the interpole type, therefore, especially lend themselves to machine-tool work where adjustable speed is required and where good speed regulation under fixed positions of the controller is necessary. They are successfully used from about 1/12 to 1,500 hp. and larger in mill and shop work.

Alternating-current Motors.—Polyphase, that is, two- and three-phase induction motors are chiefly used on alternating-current circuits in the shop. Sometimes the single-phase induction motor is used in the smaller sizes but for power work the polyphase type is nearly always employed.

The induction motor is inherently a constant-speed motor and it is largely used as such. If the rotor of an induction motor is of the phase-wound type (rather than the squirrel-cage type) with the use of auxiliary resistances in the rotor circuit it can be made to operate as an adjustable-speed motor but to a very limited extent and at the expense of efficiency and good speed regulation under various loads. Probably the greatest usefulness of the induction motor, therefore, is as a constant-speed machine. In this form, it is simple and rugged in construction, has no commutator or brushes, and when completely enclosed requires a minimum of attendance. It has a wide application in certain kinds of mill work, for example, for cement and paper mills and in reduction works. In addition to the use of auxiliary resistance in the wound rotor circuit for obtaining speed changes as well as greater torque at starting, speed changes may also be produced to a limited extent in the induction motor by the "potential," "change of poles," "change of frequency," or "cascade control" methods.

"Cascade Control" Methods.[1]—Where power is purchased from a central power station, the supply is most commonly alternating current. In such a case the induction motor will be satisfactory for most constant-speed work, and should be employed unless adjustable-speed motors are essential, thus calling for direct-current circuits. Since the usual sources of alternating-current supply are at relatively high voltages and since it is customary to utilize the power at relatively low voltages, the use of a transformer is necessary for stepping down the voltage.

Very large induction motors are often run at voltages as high as 6,600 volts, *i.e.*, direct from the line without the interposition of transformers.

The steel plant at Gary, Ind., successfully uses induction motors of 6,000 hp. on the rolls. The motors are equipped with heavy flywheels and the design is such that the peak loads are taken by the flywheels.

Generators.—Direct-current generators have reached a high stage of development in multipolar types and at lower speeds than formerly. They have been improved mechanically and there has been a transition from the belted to the direct-connected type.

The belted type is higher speed than the direct-connected, is lighter in weight for given output, and consequently is lower in first cost.

In the direct-connected type there is a saving in floor space, no belting with its dust and dirt, less noise and a saving of one bearing in moderate-sized machines as the generator is mounted on an extension of the engine shaft.

In buying a direct-connected set, if the engine is bought from one

[1] For information on these methods see "American Handbook for Electrical Engineers," John Wiley & Sons, New York, pp. 1009–1011.

manufacturer and the generator from another, it is customary for the engine manufacturer to furnish the shaft. Details are furnished the generator manufacturer and the shaft is sent to him. He then mounts the rotating member of the generator upon it.

In modern practice only the smaller units are belt-driven.

Alternating-current generators in the earlier designs were of the single-phase type, with high frequency, 125 to 133 cycles (now obsolete), and were usually of small capacity and designed for belted operation.

Rapid development has been made in polyphase types both two- and three-phase, 60 and 25 cycles.

Single-phase motors are not good in large sizes, owing to the fact that they are not self-starting, except by the use of special starting devices which in general do not afford good starting torque.

Polyphase systems are dictated from power considerations since the polyphase motor is self-starting, and gives highly satisfactory service.

The polyphase generator is superior to the single-phase in performance (regulation), cost and weight.

Distribution is cheaper with three-phase than with single-phase.

Single-phase systems are limited to small plants where the motor load is very light. A single-phase machine of 200-kw. capacity is very rare.

Lighting and power systems may both be fed from the same polyphase lines, although lighting circuits should always be kept separate from the power circuits so as to eliminate voltage fluctuations and the consequent flicker and unsteadiness usually found when lamps and motors are supplied from the same circuit.

A three-phase generator may be loaded to 58 per cent. of its total three-phase output when operated single-phase. There seems to be little reason for installing single-phase generators at the present time.

Three-phase systems are usually preferable to two-phase systems and are most generally used today. Where two-phase current is desired for special uses it may be obtained from the three-phase circuit by the Scott transformer connection.

Three-phase systems are always used for economical transmission.

Alternating-current generators have voltages which are seldom as low as 440 volts; 2,300 volts is common. Where transmission is over a fairly short distance, from say 6 to 15 miles, generators may and usually do operate at line voltages of 6,600, 11,000 or 13,200 volts. Generators are thus built to develop voltages as high as 13,200 volts.

Where transformers have been used to step up to higher line voltages for longer transmission distances, generators are commonly built at one of the following voltages: 2,300, 4,000, 6,600, 11,000, or 13,200. Alternating-current generators driven by steam turbines are now in operation with capacities as high as 35,000 kv.a.

Standard voltage for alternating-current circuits may be listed as follows: 110, 220, 440, 2,300, 6,600, 11,000, 13,200, 22,000, 33,000, 66,000, 90,000, 110,000 and 150,000. Standard frequencies in this country are 25 and 60 cycles per second with some 40-cycle installations. Arc lamps do not in general operate satisfactorily on circuits with a frequency below about 40 cycles. Incandescent lamps may be operated on 25-cycle circuits but with slightly less satisfactory results than with higher frequencies.

Twenty-five-cycle current is preferable for motors, especially so for rotary converters and synchronous motors. It is also preferable for transmission on account of better line regulation.

For general purposes as mixed lighting and power, 60-cycle current is used; or power purposes, 25-cycle current.

Exciters for Alternators.—All alternating-current generators require direct current for excitation of fields. In large stations exciters are direct-connected to prime movers. In small installations they are frequently direct-connected to or belted to the alternator itself.

Exciter plant must be absolutely reliable; hence a reserve of good capacity is required either in the form of additional units or in storage battery or both.

Capacity Required in Exciters.—For medium speed, small-sized alternators, the exciter capacity is usually about 2.5 per cent. of that of the generator. In large steam-turbine units, high-speed, it is about 0.5 per cent.

Ratings by Output.—All electrical apparatus should be rated by output and not by input. Generators, transformers, etc., should be rated by electrical output; motors, by mechanical output.

Rating in Kilowatts.—Electrical power should be expressed in kilowatts except when otherwise specified.

Apparent Power, Kilovolt-amperes.—Apparent power in alternating-current circuits should be expressed in kilovolt-amperes as distinguished from real power in kilowatts. When the power factor is 100 per cent., the apparent power in kilovolt-amperes is equal to the kilowatts.

Rated (Full-load) Current.—Is that current which with the rated terminal voltage gives the rated kilowatts or the rated kilovolt-amperes.

Determination of Rated Current.—If P be the rating or true watts, assuming a power factor of unity, and E be the full-load terminal voltage, then rated current per terminal is: $I = P/E$ in a direct-current machine or single-phase alternating-current generator.

$I = 0.58 \times P/E$ in a three-phase alternating-current generator. $I = 0.50 \times P/E$ in a two-phase alternating-current generator. If the power factor is other than unity, divide P by the power factor in per cent.

Temperature Rise.—Under regular service conditions the temperature should never be allowed to remain at a point which will result in permanent deterioration of the insulating material in the machine.

It has been recommended[1] that the following temperature rises referred to room temperatures of 40°C. be never exceeded. (These are to be considered as maximum permissible temperature rises above 40°C. allowable; manufacturers should keep within these limits.)

A. For cotton, silk, paper and similar materials, when so treated or impregnated as to increase the thermal limit, or when permanently immersed in oil, also enameled wire, 65°C.

B. Mica, asbestos and other materials capable of resisting high temperatures, in which any class A material or binder is used for structural purposes only, and may be destroyed without impairing the insulating or mechanical qualities of the insulation, 85°C.

C. Fireproof and refractory materials such as pure mica, porcelain, quartz, etc. no limit specified.

Effects of Semi and Totally Enclosing Direct-current Motors.—With semi-enclosing, at normal temperature rise, the output is reduced. Fully enclosing the motor reduces the rating still more below normal. Forced cooling raises the rating above normal.

The general plan today is to operate one large central station and locate at various centers substations for local distribution.

Irrespective of the form in which power is distributed, *i.e.*, alternating current or direct current, the power is delivered to the substations from the main generating plant (alternating-current plant usually now) as three-phase, high-voltage and at the substation it is transformed either as regards e.m.f. or in addition from alternating to direct current according to the demand.

For transmission a general figure of about 1,000 volts per mile may be used up to present limits of say 150,000 volts.

[1] See Standardization Rules of the American Institute of Electrical Engineers, edition of July 1, 1915.

CHAPTER XVII

DISTRICT HEATING

Although the discussion of heating from central stations belongs imarily to courses on "Heating and Ventilation," yet the transmis-n of steam or hot water for this purpose is so closely allied with wer generation that brief notes relating to this subject are added. The terial presented is largely from the published[1] data of Bushnell and :, and Gifford.

There are two distinct systems of central heating, steam and hot water. ntral heating as a byproduct has proved very attractive financially in ny cases and either system, steam or water, should give excellent ults if properly installed and managed and, for this reason, both sys-ns have become popular and have increased in number.

Central heating as a utility is very similar to any other business. be successful the heat must be manufactured as economically as possi-. It must be marketed economically and it must be made attractive th as to price and quality of service.

Advantages to the Public.—1. Comfort, even heat, always ready.

2. Cleanliness around the premises.

3. Reduction of labor.

4. Reduction of cartage through city streets.

5. Reduction of smoke nuisance.

6. Safety.

The Byproduct Plant.—The thermal efficiency of an electric plant is ry low. Even after the steam is generated 70 to 90 per cent. of the at is exhausted after passing through the prime mover. If this steam is hausted to the atmosphere, the waste is enormous. If it is condensed, e heat in the exhaust steam is transmitted to the cooling water and erated, either through the cooling tower or pond. At any rate, it is sted, but, of course, not so much of it is wasted in the condensing plant. owever, when an electric plant is so located that it can serve a good ating territory, it can make a kilowatt of electric energy for considerably s in conjunction with a heating plant than it can by running alone con-nsing. It is a problem that must be worked out for each situation, it, as a general rule, the revenue from the heat sales will more than pay

[1] District Heating," BUSHNELL and ORR, Heating and Ventilating Magazine Co., Y.

"Central Station Heating," GIFFORD, Heating and Ventilating Magazine Co., Y.

the coal bills after deducting from the heat income interest, maintenance and depreciation on the heating investment.

Combining these two utilities then increases the net earnings of each. It increases the load factor of the boiler plant and causes it to operate more economically because of the better load conditions. It increases the heat units utilized and sold. It should not increase the labor cost as the same crew can handle both. It increases the electrical output because it does away with isolated electric and power plants.

If the central plant can serve its patrons with heat, light and power, it is not half so difficult to get this business because, as a rule, the owners of the isolated plants can buy these commodities from the central plant for less money than they can make them themselves, but if heat is not furnished by the central plant they cannot afford to buy electricity and make their own heat, as they heat with the exhaust steam from their own plant which, they naturally figure, costs them practically nothing. This kind of business materially helps the electric plant because it increases its load and its net earnings.

In general, the combining of these two utilities is very satisfactory and advantageous to both.

Methods of Selling Heat.—The methods of charging for heat may be divided into five classes:

(*A*) Flat rates per square foot of radiation for hot-water systems.

(*B*) Flat rates per square foot of radiation for steam-heating systems.

(*C*) Flat rates per square foot of radiation, theoretically required according to the company's formulæ.

(*D*) Flat rate per year based on estimates of service requirements, or else based on estimate of what the customer would be willing to pay.

(*E*) Schedule prices based on the amount of steam used as shown by either steam or condensation meters.

Class "A"—Flat Rates for Hot-water Heating.—It can readily be seen that the price for heating during the heating season in the Southern States ought to be very much lower than in a State like Maine or Minnesota. The majority of heating plants are located in the Northern States and the variation in price would be approximately from 15 to 25 cts. per square foot of radiation, with 20 cts. as an average price in the neighborhood of Chicago, Ill.

Class "B"—Flat Rates for Steam Heating.—The rate for steam radiation is about 50 per cent. higher than that for hot-water radiation, the price varying from about 25 to 35 cts. per square foot of radiation for districts having a temperature condition approximately like that of Chicago, Ill. The difference in rates between steam and hot-water radiation is due to the fact that steam radiation usually transmits about 50 per cent. more heat per square foot of radiating surface than is transmitted by hot-water radiation.

Class "C"—Contracts Based on Theoretical Required Radiation.—Contracts are also based on a flat rate dependent on the amount of radiation required as determined by the formulæ of the company. In this form of contract the price is governed not only by the amount of radiation installed, which is the minimum, but also by the theoretical radiation required to heat the building, according to the formulæ adopted by the company.

Class "D"—Flat Contract.—Unfortunately with a great many heating companies many of the contracts are based on a flat price, which has been arrived at in a manner similar to that of a peddler in selling his wares. In other words, neither the buyer nor the seller has a very correct idea of what the service is worth, but after due discussion arrive at a price which becomes the basis of their agreement. There is always a temptation on the part of customers toward wastefulness where the service is based on a flat price per year and all formulæ used in figuring such contracts should take account of this fact. The average result in New York and Chicago shows that the consumer will use about 25 per cent. more steam when operating under flat-rate contracts, than when operating on a meter basis.

Class "E"—Contracts Based on Meter Readings.—In Class "E" the contracts are based on the amount of steam used as shown by either steam or condensation meters. This contract is used:

1. By central steam companies who furnish a house-to-house service and deliver the steam from a central plant to the curb wall of the customer.

2. By maintenance companies which operate the boiler plants in various buildings and supply steam to the owner from his own plant on a meter basis.

The result of an examination of the meter rates for steam heat for 39 cities is shown by the following table.

METER RATES FOR STEAM HEAT

	Pounds of steam per month	Cents, per 1,000 lb.		
		Maximum	Minimum	Average
1st	10,000	100	60	80.0
2d	10,000	90	50	70.0
3d	10,000	90	50	67.5
4th	10,000	87	45	65.0
5th	10,000	85	45	62.5
Next	25,000	80	43	60.0
Next	25,000	80	40	57.5
Next	50,000	73	38	52.5
Next	50,000	70	38	50.0
Next	300,000	67	36	47.5
Over	500,000	50	35	41.5

In a hot-water heating franchise the meter rate is not necessary, because there are no meters on the market. When they do come, they will probably be the heat-unit meter, and in that event 1,000,000 B.t.u. would be the logical basis of charge.

On a meter basis for steam heating, 1,000 lb. of steam is the basis of charge

On a flat-rate basis for hot water or steam heating the charge per annum or per season is either per square foot of radiation or per 1,000 cu. ft. of space heated.

When the charge per square foot of radiation is used as a basis it should be stated clearly that it is the radiation required to heat the building and not the radiation that is installed in the building. The required radiation is the amount necessary to install so that the company can guarantee to maintain a comfortable temperature.

With the flat rate per square foot basis, thermostatic control is almost essential to good operation and economical use of heat in the residences and buildings.

The flat-rate basis charge, of so much per thousand cubic feet of space, is not as equitable as the per square foot of radiation basis. For each and every 1,000 cu. ft. of contents do not require the same amount of heat, owing to the difference in location and difference in use. For instance, 1,000 cu. ft. in a corner drug store requires more heat at zero outside than does 1,000 cu. ft. in a dentist's office at zero outside. This method is not used very much at the present time. The most scientific method of charge and one that is finding universal favor in connection with public utility service is the "ready to serve" or "maximum demand" rate.

For example, assume a building that uses heat only 6 hr. out of 24 hr. in a day. This building demands a place on the line and requires the capacity, both in the mains and at the heating station, to supply its maximum demand. The heating company must be ready to serve this building at any time it requires service. In return for that required condition, the heating company only receives pay for 6 hr. service, while the building using heat for 18 hr. pays the company three times the revenue paid by the other building.

It is also the case with churches or auditoriums where heat is used only two or three days out of the week. The heating company must reserve capacities for these buildings, but in return receives only about one-sixth the revenue it would receive from other buildings requiring the capacities that these churches, etc., require. It is obvious, therefore, that a rate based on the number of pounds of steam condensed is far from equitable.

It is here that the "maximum demand" or "ready to serve" rate shows its strong points as an equitable rate. As stated before, this rate is based on the cost of the service to the utility, plus an equitable profit.

In making a rate for service per season it is desirable to know the percentage of heat used in any one month during the heating season for two reasons: First, the collections are often made monthly; second, it sometimes happens that a credit or debit is given for certain months during the season, when the service is started or stopped in mid-season. The following table will be useful in this connection.

HEAT CONSUMPTION TABLE

	Per cent.
Heat used up to Oct. 31	6
From Oct. 31 to Nov. 30	12
From Nov. 30 to Dec. 31	18
From Dec. 31 to Jan. 31	21
From Jan. 31 to Feb. 28	19
From Feb. 28 to Mar. 31	13
From Mar. 31 to Apr. 30	8
From Apr. 30 to May 15	3
	100

The following division is used extensively in flat rate contracts and is easily remembered by the consumers:

5 per cent. of contract price payable Oct. 1.
15 per cent. of contract price payable Nov. 1.
20 per cent. of contract price payable Dec. 1.
20 per cent. of contract price payable Jan. 1.
20 per cent. of contract price payable Feb. 1.
15 per cent. of contract price payable Mar. 1.
5 per cent. of contract price payable Apr. 1.

Before leaving the subject of rates, it might be well to call attention to the two-rate system, which has been frequently advocated, viz., the adoption of a primary charge based on the theoretical amount of radiation connected and a secondary charge based on the meter readings. This kind of rate is perhaps more thoroughly sound than the single sliding-schedule, due to the fact that the primary rate can be made to closely approximate the investment charge, while the secondary rate can be used on the operating costs. The chief objection to this rate is that it requires that the theoretical radiation for each customer be figured from a basic formula and such estimates of theoretical radiation required are more open to controversy than the reading of a satisfactory meter. Another point in favor of the simple sliding-schedule based on meter readings is that it is more easy to explain this method of charging to the customer.

While it is very possible that the two-rate system will be the future basis for the sale of steam, just as it is already to a large extent the basis for the sale of electricity, yet it is questionable as to whether the time for this change has arrived.

One two-rate system in use in an eastern city is applied as follows:

(*a*) Reduce the cubic feet of heated contents to a cube of equal contents.

(*b*) Base the rate on the area of one side of this cube and on the actual steam consumption.

For the city in question the rate is 1 ct. per square foot of the equivalent cube plus 30 cts. per 1,000 lb. of steam.

Take a house of approximately 20,000 cu. ft. of heated contents. Equivalent cube = 27 × 27 × 27. Fixed charge each month of heating season (8 months) = 27 × 27 × $0.01 = $7.29.

If the steam consumption for a month were 20,000 lb., the bill for the month would be

Fixed charge =	$7.29
20 × 30 cts. =	6.00
	$13.29

If the consumption for the month were 50,000 lb. then the bill would be

Fixed charge =	$7.29
50 × 30 cts. =	15.00
	$22.29

Cost of Exhaust Steam Heating.—The Wisconsin Railroad Commission, in its reports, vol. 2, page 302, states that, with coal at $2 per ton, exhaust steam could be sold at 50 cts. per 1,000 lb. safely; that is, with a secure profit, but it also states that where live steam is sold there would be small profits unless large quantities of steam were sold.

The engineer of a company of recognized standing states that the charge which his company makes to the different buildings and departments for the use of exhaust steam for heating is at the rate of 2 cts. per month per square foot of radiating surface for 6 months of the year.

He finds in practice that when using exhaust steam 1 lb. will supply about 3 sq. ft. of radiation with steam for 1 hr. He figures that if 1 lb. of coal will produce 8 lb. of steam, it will supply about 25 sq. ft. of radiation for 1 hr., and when coal costs $2 per ton the fuel necessary to supply 1 sq. ft. of radiation for one month, making proper allowance for the value of the steam used in the engine, will cost about 1 ct., assuming that steam is supplied to the radiator for only 10 or 12 hr. per day.

Also assuming that the amount of boiler-room expense will just about equal the fuel cost, he estimates the cost for exhaust steam as 2 cts. per square foot per month during the months when heat is necessary. It is very doubtful whether many heating plants can be operated for heating alone and supply live steam at a cost of 2 cts. or less per square foot per month.

The normal consumption of steam in radiators per season under ordinary conditions in a climate similar to that of Chicago, Ill., runs from 600 to 800 lb. of steam per square foot where the radiating surface is properly proportioned. There are buildings, however, which cannot be depended upon to run very close to this average. Occasionally the consumption will be very much above this amount in one building and in another building it will be very much below, the amount used running all the way from 300 to 3,000 lb. per square foot of radiating surface during the heating season.

The following summaries present interesting data showing the variations in steam consumption in different latitudes.

Class of buildings—office buildings, retail stores, residences, saloons, hotels, apartments, garages, light manufacturing buildings, wholesale stores, clubs, schools, churches, offices, banks, lodges, factories, theatres, restaurants, post offices, Y. M. C. A.'s, halls, telephone companies, hospitals, city halls, court houses, etc.

Large City in the Middle West

Mean Temperature during Heating Season 41°F.

Number of consumers	Cubic feet of space	Square feet of radiation	Total condensation per season, pounds	Season's average per 1,000 cu. ft. of space, pounds	Season's average per sq. ft. radiation, pounds
162	20,505,000	229,842	117,196,000	5,715	510
Eastern City					
Mean Temperature during Heating Season 41.3°F.					
187	7,258,293	94,993	63,249,000	8,714	665
Large Southern City					
Mean Temperature during Heating Season 54.7°F.					
149	24,546,000	194,702	84,711,000	3,451	435

Steam Consumption of Various Classes of Buildings in a City of the Middle West, Indicating the Economy Effected by Meter Rates

Meter rate					
39	7,103,028	65,132	38,012,000	5,352	584
Flat rate					
95	7,859,357	78,062	74,318,000	9,456	951

Estimating Miscellaneous Steam Requirements in Large Buildings.—In addition to heating service the district-heating company may be called upon to supply steam for many other purposes. The owner of the modern first-class building is obliged to provide the highest quality of service for his tenants, and this is accomplished only by the installation of a system of auxiliary apparatus of various kinds, many of which are operated by steam. Among these may be included the following:

1. Hot-water heaters, supplying heated water for industrial or manufacturing purposes, or for domestic uses, such as scrubbing and lavatories.
2. Vacuum pumps, used in connection with steam-heating systems for removing air and condensation from the radiation.
3. Ejectors used in a manner similar to vacuum pumps.
4. House pumps used for elevating the domestic water supply to the roof of the building where it is stored in a tank.
5. Boiler-feed pumps.
6. Steam-hydraulic elevator pumps.
7. Direct-steam elevator engines.
8. Fire-pumps.
9. Air compressors.
   (*a*) General use.
   (*b*) Sewer-ejector system.
   (*c*) For pressure-tanks on hydraulic system.
10. Steam syphons and jets.
11. Refrigerating machinery.
    (*a*) Compression systems.
    (*b*) Absorption systems.
12. Brine pumps or other auxiliary refrigerating apparatus.
13. Drinking-water pumps.
14. Stoker engines.
15. Ventilating fan-engines.
16. Warming and cooking apparatus for restaurants.
17. Laundry apparatus.
18. Miscellaneous industrial uses, varying with the class of building and tenants.

To determine with any degree of certainty just how extensive the use of steam for these purposes will be in each building, usually requires a specialist in this particular line of engineering—one who is enabled to draw from experience and observation for verification of estimates. Whenever possible, it is needless to say one should be guided by comparison with buildings already supplied with approximately similar service.

The cost of heating service is made up of the following items which should be figured on the basis of 1,000 lb. of steam generated or a multiple of that quantity.

1. Fixed charges based on investment in:

Building,	Dynamos,
Boilers,	Furnaces,
Piping,	Engines.

and various accessories for the above.

(a) Amortization.
(b) Obsolescence.
(c) Interest.
(d) Taxes.
(e) Rental value of space.
(f) Marginal charge for diversion of capital.

Operating costs:

Salaries:
- Chief engineer.
- Assistant engineers.
- Firemen.
- Coal-passers.
- Oilers.
- Electricians.
- Steam-fitters.
- Boiler-washers.
- Elevator repair men.
- Helpers.
- Engineer's clerk.
- Office labor for metering and billing.
- Employer's liability insurance and salaries paid to injured employees when off duty.

Also a portion of the time of the management used in buying supplies and looking after the operating organization.

Supplies:
- Fuel.
- Transportation of ashes.
- Oil, waste, water.
- Shovels, fire-tools.
- Electricity for lighting and power in boiler room.
- Miscellaneous supplies and expenses.

All of the above are direct costs which are directly chargeable to the cost of operating a plant. In addition to the above costs, there are other costs which might be termed indirect charges which often come as a result of power-plant operation.

1. *Throwover Switch Service from Central-station Service.*—As a rule, the cost per kilowatt-hour for throwover switch service is greater than the rate where complete service is furnished, and often a minimum bill is required in addition to the higher rate.

2. *Danger of Breakdown in the Service and Consequent Loss if Throwover Switch is not Installed.*

3. *Losses on Account of Decreased Rental Value of the Building.*—The majority of isolated plants operated with high-speed engines shows a marked fluctuating quality in the light. This is usually increased at irregular intervals where high-speed electric elevators are operating on

the same plant. It is also frequently found that in the summertime the space directly above the boiler is hard to rent on account of the heat coming up through the floor from the engine room below. There is also the damage and annoyance caused by vibration in the building.

4. *Losses of Time on Account of Obstruction of Entrances by Coal Teams.*—Some of the firms which have discontinued the use of their own plants and gone on central-station service have been particularly desirous of getting the steam service also in order that they might discontinue the delivery of coal to their buildings, and thereby be able to receive and deliver goods without any interference with coal teams.

5. *Losses on Account of Smoke Fines, or Dirt in the Building, due to Operation of the Boilers.*—In large Western cities where soft coal is used, there has been an active campaign started to prevent the emission of smoke, and a number of these cities have laws imposing fines on the owners of smoky chimneys.

6. *Losses on Account of Strikes, due to Labor Troubles.*—If the above costs of operation are carefully tabulated and are based not on the theoretical economy of apparatus operating at maximum load when new and under special conditions, but on the average operating economy as found in plants, they will show a substantial saving by the use of central-station service, providing the rates for central-station service correspond with those recently made in many of our large cities.

Heating Station.—Gifford states that in the heating station we have the same general subdivision in our subject, (1) steam and (2) water. Steam systems divided into (*a*) vacuum and (*b*) pressure, and water systems are divided into (*a*) open and (*b*) closed systems.

Steam System.—The vacuum system has a low pressure on the steam main and a vacuum or suction on the return main. Pumps are used to create the vacuum and thereby to cause the return water to come back to the plant.

Its operation is as follows: The steam is generated in the boiler and passes either through the engines, after which the oil is extracted, or direct to the pipe line. Through the pipe line it goes to the buildings. It is there condensed and the condensation returns to the plant via the return line and is usually emptied into a storage tank. Here it is stored until the boilers need it. It passes from the tank to the feed-water heater, then through the economizer, if one is used, and then into the boiler again.

The pressure system is like the above except that there is no return and, consequently, no vacuum pump. The pressure carried on this system is optional with the designer, but better practice seems to be to keep the pressure as low as possible, especially if the engines exhaust into the heating mains. Some engineers design, in straight-fuel-burning plants, high-pressure systems and reduce the pressure at the service of each building or in the high-pressure feeder lines, but this practice has never been very popular and its advantages have never been proven. The modern practice seems to be a low-pressure system (3 to 10 lb.) with high-pressure feeders to be used as such only when necessary.

The operation of a pressure system is very simple. The steam is generated in the boilers and goes from there to the engines and then to the pipe line, if an exhaust steam plant. If not, the steam goes direct to the pipe line and through the pipe line to the buildings. Here it is condensed and the condensed water cooled as much as possible and then dumped into the sewer.

The pressure system is used more extensively than the vacuum system.

The vacuum system allows a somewhat lower pressure on the engines and also furnishes the return water for reuse, which is a benefit sometimes.

Water Systems.—The open system of hot-water heating gets it name from the fact that the system is open to the atmosphere at one point. In this system we find the open heater or com-mingler used. The operation is as follows: The water returns from the pipe line and passes through a relief valve to the open heater or com-mingler where it is reheated to as high a degree as there is exhaust or live steam to heat it. It then goes through the circulating pump and is forced, if sufficiently heated, out into the line again. If not sufficiently hot it is forced through a closed high-pressure condenser or heater, or through a circulating boiler where it is reheated more, before going out into the pipe line and to the buildings. After it goes through the buildings it returns to the plant again.

In this system the steam and circulating water mix in the open heater or the com-mingler and water is, therefore, added to the system. Consequently, it is necessary to supply some means of relief which is done by placing a relief valve on the return line; when the pressure exceeds a certain point, which it will do if water is added, the valve will open and discharge into the storage tank. This can also be accomplished by floating the storage tank on the return line and letting the overflow go direct to the tank, but this arrangement does not allow any appreciable change in the return pressure, which change is sometimes desirable.

The advantage of this system is that all the heat in the steam is transferred into the circulating water, this giving us a high efficiency in the transmission of the heat, but it requires more energy in the pumping of the water due to the reduction in the pressure which is necessary in order to mix the steam and water and liberate the air.

Liberating or extracting the air is simple in an open heater because the pressure on the heater is the same as the atmosphere and a vent open to the air will carry off the air. In a com-mingler where from 3 in. to 16 in. of vacuum is maintained on the exhaust steam line it is not so simple, but this can be accomplished by means of tanks which work very well.

The difference in operation between a com-mingler and an open heater is that the com-mingler will create a vacuum on the engine exhaust and the open heater will not. However, the com-mingler will operate at atmospheric pressure if it is desired.

The open system throws all the condensed steam into the heating mains, which is an advantage in one way, in that it keeps the heating mains full of good water and is a disadvantage in that it takes this water away from the boiler. But with a tight pipe line the water discharged from the overflow of the pipe line soon becomes fairly good because all make-up water is condensation and, consequently, the water for the boilers, which should be taken from the storage tank, will soon be diluted with sufficient good water to make the feed water fairly good, so that this disadvantage is not troublesome.

The closed system of hot-water heating operates as follows: The water returns to the plant and goes through the circulating pumps. In this way the pumps handle the coolest water, which is some advantage. Then it goes through the exhaust steam condenser or heater and then, if not sufficiently warm, it goes through the high-pressure steam heater or the circulating or reheater boilers. Sometimes the circulating water is forced through economizers. This is a very good arrangement. After the water is reheated sufficiently, it is forced out through the pipe line through the buildings and back to the plant.

The closed system operates on less power requirements in its circulation of water. The fact that it does not mix the steam with the water insures good feed water for the boilers. The heat transmission of the condensers or closed heaters in this system is not as great as in the open heater or com-mingler in the open system. All water entering the system should be treated so as to take out as many of the impurities as possible and thus keep the line in good shape by putting in only good water.

Which system to adopt is a question that can be decided only after local conditions are known. It is a difficult matter sometimes to determine—and generalities in this connection are misleading. If a steam plant is decided upon, it is not difficult to figure out which system, vacuum or pressure, will be the most economical. If a water plant has been chosen, it is more difficult to decide whether an open or a closed system will be the better. If a vacuum is desired on the engines, an open system with a com-mingler is a desirable and logical choice. If there are no engines to be considered as, for instance, in a straight-fuel-burning plant, a closed system answers the purpose better than an open system because it requires less power to handle the circulation. A closed system will create a vacuum if the condenser surface is sufficiently large and the water mperature is not carried too high and if there is not too much steam to ndense. As a general rule, if the hot-water plant is a byproduct to an ectric plant, an open system is the better. If not a byproduct plant, e closed system is, perhaps, preferable.

After the system has been decided upon, the details of the design can determined. In deciding upon these points capacity and efficiency e the main features to be considered. Durability is also important and is the expense of maintenance.

CHAPTER XVIII

THE POWER PLANT OF THE TALL OFFICE BUILDING[1]

Probably the best illustration of compact power installations to meet the heating, ventilating, lighting, power and sanitary demands of a good-sized town or small city is found in the plants of tall office buildings and modern hotels.

Data from 17 such plants show the cost of electric current to be made up of labor, coal and handling ashes, water, lamps, oil and supplies, repairs, central station service where used for periods of minimum consumption to allow shutting down of the plant, and interest and depreciation.

Roughly these figures average: labor, 1/3; coal, 1/3; interest and depreciation, 1/10 or more; and the sum giving the cost of power as made up of the minor items.

The mean load throughout the day is usually about 50 per cent. of the full load. The maximum load is required only an hour or two in the afternoon during the winter months. It is customary to allow about 50 per cent. reserve over the estimated peak loads in designing the boiler plant.

Division of the Load.—The average power required during the different portions of the year is shown in the following table:

	Per cent. of total	Lights, kw.	Power, kw.	Total, kw.
Absolute peak load	70	214	20	234
Average peak and running load, dark days	60	184	20	203
Average peak load for 8 months	30	92	20	112
Average day load for 8 months	30	92	20	112
Average day load for 6 months	20	62	20	82
Average low days for 12 months	16	49	20	69
Average nights, Sundays and holidays	..	...	..	50

In addition to the above there is usually an increase of 10 per cent. over the above running loads on account of the special needs of tenants.

Under ordinary conditions it is customary to allow 1.6 hp. in engines for each kilowatt output of the generator and 1.8 hp. in boilers for each kilowatt in generators.

Selection of Plant.—In accordance with the above the following main plant would be selected, which gives elasticity in its working and will take care of the following conditions:

[1] Summarized from "The Power Plant of the Tall Office Building" by J. H. Wells, *Transactions* A.S.M.E., vol. 25, p. 685.

1. Maximum load 70 per cent. of total connected load + 10 per cent. = 257 kw.

2. Average day load for 8 months 30 per cent. of total connected load + 10 per cent. = 123 kw.

To operate under these conditions the following plant will meet the requirements.

Two generators of 125-kw. capacity each, either of which will carry the average peak running load, or both connected will carry the absolute peak loads which are on for short isolated periods only.

One generator of 100-kw. capacity to carry early running and low average loads for 12 months.

One generator of 50-kw. capacity as an auxiliary unit for nights, holidays, Sundays and odd times.

Such a plant in operation was called upon for the following:

Kilowatt-hours of lighting load

January	33,670	May	19,301	September	17,688
February	25,388	June	18,200	October	25,542
March	24,462	July	16,600	November	33,696
April	19,950	August	17,928	December	40,425

Assuming, therefore, a total output through the busbars of 431,050 kw.-hr. per year, the engines would generate

$$1.6 \times 431,050 = 689,680 \text{ hp.-hr.}$$

and the boilers would generate

$$1.8 \times 431,060 = 775,890 \text{ hp.-hr.}$$

Assuming an engine guarantee of 24 lb. of steam per hp.-hr. and 8 lb. of water per pound of coal, then the total coal will be 1,164 tons and the water required will be 297,942 cu. ft.

It is safe to assume that only 50 per cent. of the water is wasted and the remainder returned to the boilers.

Taking cost of coal at $3.75 per ton; water at 10 cts. per 100 cu. ft. (New York prices); oil and waste at $4.60; interest and depreciation (if one-half be charged to this account), $3,000; ash removal, $218.25 (5 per cent. of cost of coal); and charging one-half the cost of the force in the fire and engine rooms to this account, or $2,500, makes the total cost of operating the electric plant $10,632, or approximately 2.5 cts. per kw.-hr.

Estimating Boiler Capacity Required.—Hubbard gives the following methods of computing the boiler capacity required in office buildings.

Heating.—Boiler power for heating is usually obtained from the amount of radiating surface to be supplied, and for all practical purposes the following ratios may be used, in which it is assumed that 1 boiler hp. will supply 130 sq. ft. of direct cast-iron radiation, 100 sq. ft. of direct wrought-iron pipe coils, 50 sq. ft. of indirect cast-iron radiation. Boiler power computed in this manner should be increased about 5 per cent. for losses due to radiation from steam and return mains.

Ventilation.—Boiler power required for ventilation is based on the volume of air to be supplied. This may be found by the rule that the horsepower is equal to the cubic feet of air to be warmed per hour from zero to 70°, multiplied by 1.3 and divided by 33,000.

Power for Lights.—In finding the boiler power for lighting, first determine the electrical energy to be supplied at the lamps, then find the indicated horsepower of the engines necessary to produce this, and then compute the probable quantity of steam required from the type of engine to be used.

In computing the approximate electrical horsepower at the lamps it may be assumed that in general for offices, assembly halls, etc., approximately 1.25 watts will be required per square foot of working plane for good lighting.

The efficiency of a first-class generating set, including the losses in transmission, may be taken as about 75 per cent. when located in or near the building to be lighted, so that the electrical horsepower necessary to supply the lamps divided by 0.75 will give the indicated horsepower of the engines.

The total weight of steam required, in pounds per hour, divided by 30, will give the approximate boiler horsepower.

Driving Fans.—Power for driving the fan motors may be included with the power for lighting, by assuming 1 hp. of electrical energy delivered to the motors for each 2,000 cu. ft. of air to be removed by the fans per minute.

Hot Water.—The boiler power for hot-water heating may be determined by multiplying the number of gallons of water to be heated per hour by the increase in temperature and by 8.3, then divide by 33,000 to reduce to horsepower. The temperature increase may be taken as 140° under average conditions.

Elevators.—The horsepower required for raising an elevator is found by multiplying the sum of the live load, which averages about 70 lb. per square foot of floor space in the car, and the unbalanced weight of car, which may be taken as approximately 25 lb. per square foot of floor space in hydraulic elevators, and 0 for drum and duplex electric elevators, by the speed in feet per minute, average 400 to 600. Divide this product by the efficiency, average about 0.6, and divide again by 33,000 to reduce to horsepower.

Allowing for stops and time when the elevators are idle it is customary to consider that each elevator is in operation 0.7 of the time. As hydraulic elevators are not counterweighted up to their full weight, they descend by gravity, so power is required only on the upward trip.

The above rule is for a continuous upward movement, hence, if the elevator is in operation only 0.7 of the time, and one-half the time that it is in actual operation is occupied in downward trips, requiring no power, the results found by the above calculation should be multiplied by 0.7 times 0.5 when considering a hydraulic elevator. Substituting the average values as given above shows the required horsepower for each square foot of floor space in the car to be

$$(70 + 25) \times 400 \times 0.7 \times 0.5 \div 0.6 \times 33{,}000 = 0.7$$

Hence, the total square foot floor area of the elevator cars multiplied by 0.7 will give the horsepower to be delivered by the pumps.

The necessary boiler power will depend upon the type of pump used. The following table gives the average steam consumption in pounds per delivered horsepower per hour for different types of duplex pumps.

Type of pump	Rate of steam consumption, pounds per horsepower-hour
Simple non-condensing	150
Compound non-condensing	85
Triple non-condensing	45
High-duty non-condensing	35

The total horsepower required multiplied by the rate of steam consumption divided by 30 will give the boiler horsepower.

For electric elevators the foot-pounds per minute are readily determined and hence the horsepower for each square foot of floor surface in the cars.

Assuming a combined efficiency of 0.65 for the engine, generator and motor, the indicated horsepower of engine square feet of floor surface in the cars is determined and this multiplied by the total floor space, in square feet, will give the total indicated horsepower of the engines. From this point on the method of obtaining the boiler horsepower is the same as already described in the case of electric lighting.

Refrigeration.—The capacity of a refrigerating plant is commonly expressed in two ways: "ice-melting effect" and "ice making." For example, a 20-ton machine will produce the same cooling effect in 24 hr. as the melting of 20 tons of ice, or in other words, will extract the same amount of heat from the brine as would be required to melt 20 tons of ice into water at a temperature of 32°.

Theoretically, the extraction of this amount of heat from 20 tons of water, at an initial temperature of 32° should change into ice; but in practice there are various losses not present in the simple process of cooling, so that it is customary to allow for twice the boiler power per ton for ice making as for the process of cooling or ice-melting effect. The indicated horsepower required per ton of refrigeration depends upon the suction and condenser pressure, which in turn are governed by the temperature and amount of the condensing water used.

Under conditions where condensing water must be obtained at average city prices the most economical results are obtained with suction pressures ranging from 20 to 30 lb. and condenser pressures of 140 to 150 lb. Under these conditions one i.hp. in the steam cylinder will produce about 60 lb. of ice-melting effect per hour, or 0.75 ton per 24 hr. This will, of course, vary somewhat with the range of pressures and also with the size and type of machine, but in the absence of more exact data, may be used for approximate results. Another method in common use is to provide 1.5 i.hp. per ton of refrigeration, which is slightly more than the previous case. Knowing the indicated horsepower of the compressor, the probable steam consumption can be determined for the particular type of engine used.

Comparative Costs of Private and Central Station Heating and Power.—To illustrate the comparative costs of private plants *vs.* central station heating and power, Bushnell and Orr present[1] the following figures from a building recently analyzed in Chicago. This building is a large office building about 200 ft. square and 21 stories in height.

It has a court in the center above the first floor 73 ft. square. The original estimate of the steam consumption based on formulæ was 63,200,000 lb. The actual consumption during the year 1913 as shown by meters was in round numbers 64,300,000 or about 1,100,000 lb. over the estimate.

As the steam consumption in any building will vary ordinarily a much larger percentage from season to season, the estimate given may be considered fairly accurate. The original estimate for consumption of electricity was 1,250,000 kw.-hr. The consumption in 1913 was 1,100,000 kw.-hr. If a plant had been installed in the building, the consumption would probably have been about 50,000 kw.-hr. more, and as the building is not quite rented a complete rental of the building would probably bring the current consumption very nearly up to the estimate.

The actual consumption for this building was in round numbers 500,000 kw.-hr. for tenants' lighting, 150,000 kw.-hr. for public lighting and 450,000 kw.-hr. for power, of which about three-quarters was consumed by the elevator equipment. Assuming a price for electricity of $2\frac{1}{2}$ cts. per kw.-hr. from the central-station service and a price of 40 cts. per 1,000 lb. for steam on central-station service, it is very easy to figure the cost of central-station service on this basis. Let us assume that the building will be fully rented and that the total consumption is

[1] Bushnell and Orr, "District Heating."

1,150,000 kw.-hr. We will also assume that the building purchases its entire requirements both for steam and electricity and retails the electricity to its own tenants. The total bills for the building would be:

1,150,000 kw.-hr. at 2½ cts. per kilowatt-hour	$28,750
64,300,000 lb. of steam at 40 cts. per 1,000 lb.	25,720
Total	$54,470

In figuring the cost of isolated plant service, it will be necessary to add the cost of electricity for lights in engine and boiler rooms, and also the cost for ventilating same. Assuming, therefore, that this amounts to 50,000 kw.-hr. per year, the total electricity used by the plant would be 1,150,000 kw.-hr. plus 50,000 kw.-hr. or 1,200,000 kw.-hr. per year. The average steam consumption in office building plants as shown by a number of tests taken on typical installations is about 60 lb. of steam per kilowatt-hour throughout the year. While the above would represent average conditions, in this comparison it would be better to assume 50 lb. since in a large building such as this, it would be possible to get an economy above the average.

1,200,000 kw.-hr. of electricity at 50 lb. per kilowatt-hour would require 60,000,000 lb. of steam per year. It is fair to assume that about 40 per cent. of this would be saved for heating by utilizing the exhaust from the engines. This would leave a net steam consumption of 60 per cent. of 60,000,000 or 36,000,000 lb. It has been shown by meter readings that the heating requirements of the buildings are 64,300,000 lb. of steam. Adding together the steam required for electricity and the steam for heating, gives a total of 100,300,000 lb. or in round numbers 100,000,000 lb. of steam per annum. The average evaporation in this plant runs about 5 lb. of steam per pound of coal. If a power plant were operated all summer long the average evaporation would be somewhat higher, say 5½ lb. of steam per pound of coal. On the basis of 100,000,000 lb. of steam, the annual coal consumption would be 18,181,818 lb., or in round figures 9,000 tons. On this basis the operating expenses would be as follows:

SUPPLIES

9,000 tons of coal at $2.75 per ton	$24,750
Ash removal, 6 per cent.	1,485
Water, for steam supply, washing out boiler and engine room, etc.	1,000
Oil, waste and packing	1,200
Tools and miscellaneous supplies	1,200
Boiler and fire insurance	60
Total	$29,695

LABOR

Chief engineer	$3,000	
Assistant to chief engineer	1,500	
Three watch engineers	3,600	
Two oilers	1,920	
Engineer's clerk	480	
Three firemen at $840	2,520	
Two ashmen at $720	1,440	
Liability insurance and losses from sickness among employees	1,000	
Time of office, including manager's time for supervising	1,000	16,460
Total operating expenses		$46,155

In addition to the operating costs we must include the:

Fixed Charges.—To take care of this building which has an aggregate installation of about 15,000 50-watt lamps, 200 hp. in general power and 600 hp. in elevator power, or a total connected equipment of about 1,800 hp., it will be necessary to install a plant of about 1,200 kw. which would cost complete at $50 per kilowatt about $60,000. The plant would also require space of upward of 6,000 sq. ft. On the above basis the fixed charges would be as follows:

Amortization at 3 per cent	$1,800
Obsolescence at 5 per cent	3,000
Interest at 6 per cent	3,600
Repairs at 2 per cent	1,200
Taxes at 1 per cent	600
Rental value of space at 50 cts. per square foot	3,000
Marginal charge for diversion of capital at 5 per cent	3,000
Total	$16,200

Summarizing the above gives:

Operating charges	$46,155
Fixed charges	16,200
Total	$62,355

It will be noted that the cost for labor to take care of the elevators, electric fans, etc., as well as the radiation has been omitted from both estimates as they are practically equal in both propositions. Comparing this with the above cost of central-station operation, we find a saving of about $8,000 per year. As a matter of fact the central station costs in Chicago are slightly under these figures. If the price for electricity, however, were 4 cts. per kilowatt-hour and the price of steam 50 cts. per 1,000 lb., the situation would be reversed, and there would be a saving of

000 in the operation of an isolated plant. In other words, the t determined by the cost of isolated plant operation, but by the :d by the central-station company.

ove figures are given as average figures and may be found to be ower in different localities and in different plants. The fact of the largest buildings in Chicago now operating plants are considerably higher expense than that assumed in this esti- s to show that the estimated cost of isolated plant service is re.

PROBLEMS

-story office building occupies a ground area of 15,000 sq. ft. and has a ty of 2,000,000 cu. ft. There is 1 sq. ft. of direct cast-iron radiating sur- l per 90 cu. ft. of space. The elevator equipment is four 10-passenger 0 ft. per minute; one 1-ton freight elevator, 400 ft. per minute. All motor-driven.

floor is ventilated, 10 changes of air being provided per hour. Hot-water rs will provide 1,000 gal. of water per hour at 180°F.

nes are simple high-speed, arranged to run condensing in summer and to the heating system in winter.

e the capacity of generating equipment required for the building. Will apacity be required for heating? If so, what is total capacity required.

CHAPTER XIX

THE POWER PLANT OF THE STEAM LOCOMOTIVE

The most compact steam power plant in daily commercial use is found in the locomotive.

Horsepower.—The indicated horsepower of the locomotive may be computed as for any steam engine, but it is sometimes more convenient to use the following formula:

Let p = mean effective pressure in pounds per square inch.
d = diameter of cylinder in inches.
S = length of stroke in inches.
M = speed in miles per hour.
D = diameter of driving wheel in inches.

Then for two cylinders

$$\text{i.hp.} = \frac{pd^2SM}{375D}.$$

The indicated horsepower of some of the latest type compound locomotives runs as high as 2,500.

Efficiency.—The power required to overcome the friction of the moving parts of the locomotive and to drive the locomotive and tender varies from 10 to 30 per cent. of the developed power.

Tractive Force.—*Simple Locomotives.*—

Let F = indicated tractive force in pounds.
p = mean effective pressure in the cylinder in pounds per square inch.
S = stroke of piston in inches.
d = diameter of cylinders in inches.
D = diameter of driving wheels in inches.

Then

$$F = \frac{4\pi d^2pS}{4\pi D} = \frac{d^2pS}{D}.$$

If the drop in pressure of the steam due to expansion, friction and wire drawing be taken into account, then the formula becomes

$$\text{Actual tractive force} = \frac{0.8Pd^2S}{D}$$

in which P = boiler pressure in pounds per square inch.

Compound Locomotives.—The Baldwin Locomotive Works formulæ for compound locomotives of the Vauclain four-cylinder type are

$$F = \frac{C^2S \times 0.67P}{D} + \frac{c^2S \times 0.25P}{D}$$

in which C = diameter of high-pressure cylinder in inches.
c = diameter of low-pressure cylinder in inches.

For a two-cylinder cross-compound the formula is simply

$$F = \frac{C^2S \times 0.67P}{D}$$

The formulæ apply for speeds not over 10 miles per hour above which the tractive power rapidly falls off. The limit of hauling capacity of a locomotive is usually from one-fifth to one-fourth of the weight on the drivers.

Drawbar Pull.—The drawbar pull or tractive force of a locomotive is the force exerted at the drawbar or connection to the train as indicated on a dynamometer. This force is limited by the weight on the driving wheels and by the power of the engine. The drawbar pull at that point where the driving wheels begin to slip is known as the adhesion of the locomotive.

Adhesion varies with the cœfficient of friction between steel and steel under given track conditions. The use of sand under the drivers is to increase the cœfficient of friction.

VALUES OF COEFFICIENT OF FRICTION

Good conditions, dry rail	f = 0.20 to 0.25
Maximum with sand	f = 0.33
Wet sloppy rail as in fog	f = 0.15
Worst condition	f = 0.125

Tractive force = weight on drivers × f.

Under good conditions the drawbar pull necessary to haul 1 ton of 2,000 lb. varies from 6 to 8 lb. on level track and increases with curves and grades.

Two formulæ in general use for determining the resistance are

$R = 2 + \frac{V}{4}$ (*Engineering News* formula).

$R = 3 + \frac{V}{6}$ (Baldwin Locomotive Works formula).

in which R = resistance in pounds per ton (2,000 lb.) on straight, level track,
and V = velocity in miles per hour.

The increased resistance due to grade is as follows:

If the grade be 1 ft. per mile, the pull required to lift 2,000 lb. will be $\frac{2,000}{5,280} = 0.3788$ lb.

Total resistance due to grade in pounds per ton (2,000 lb.) = 0.3788 × rise in feet per mile.

If the grade is expressed in per cent. the resistance in pounds per ton (2,000 lb.) will reduce to 20 lb. for 1 per cent. grade.

The resistance due to curves is not easily determined. G. R. Henderson in "Locomotive Operation" estimates this resistance at 0.7 lb. per ton (2,000 lb.) per degree of curve.

(Degree of curve = the angle at the center subtended by a chord of 100 ft.)

Resistance in pounds per ton = 0.7 c.
in which c = number degrees of curve.

This value is greater for locomotives, often being as high as 1.4 c.

Resistance due to acceleration $= 70 \frac{V^2_2 - V^2_1}{S}$
in which V_1 and V_2 = velocities in miles per hour and S = distance in feet.

Increased Economy with Increase of Pressure.—Tests reported (*Bulletin* No. 26, University of Illinois Experiment Station) show the following increase in economy with increase in boiler pressure:

Boiler pressure, lb. per sq. in.	120	140	160	180	200	220	240
Steam per i.hp.-hr., lb.	29.1	27.7	26.6	26.0	25.5	25.1	24.7
Coal per i.hp-hr., lb.	4.0	3.8	3.6	3.5	3.4	3.4	3.3

Effect of Speed on Average Steam Pressure.—C. H. Quereau points out (*Engineering News*, Mar. 8, 1894) that the mean steam pressure (and consequently the power of the engine) decreases as the speed of the locomotive increases. He gives the following figures:

Miles per hour	46	51	51	53	54	57	60	66
R.p.m.	224	248	248	258	263	277	292	321
Pressure, lb. sq. in.	51.5	44.0	47.3	43.0	41.3	42.5	37.3	36.3

Two-cylinder compound			Single-expansion		
Revolutions	Miles per hour	Water per i.hp. per hour	Revolutions	Miles per hour	Water per i.hp. per hour
100 to 150	21 to 31	18.33 lb.	151	31	21.70
150 to 200	31 to 41	18.9 lb.	219	45	20.91
200 to 250	41 to 51	19.7 lb.	253	52	20.52
250 to 275	51 to 56	21.4 lb.	307	63	20.23
			321	66	20.01

Effect of Speed on Steam Consumption.—Mr. Quereau also gives in the ıe article the foregoing table relating to the variation of steam con- ıption with speed.

Depreciation of Locomotives.—Kent quotes the Baldwin Locomotive rks as suggesting that for the first 5 years the full second-hand value he locomotive (75 per cent. of first cost) be taken; for the second 5 rs 85 per cent. of this value; for the third 5 years, 70 per cent.; after years, 50 per cent. of the second-hand value; and after 20 years, and ong as the engine remains in use, 25 per cent. of the first cost.

Jse of Superheated Steam.—Superheated steam is now quite gen- y used in locomotive practice. Its use has resulted in increased m economy and less trouble from water of condensation in the cylin- . The saving in water consumption per horsepower-hour is reported e some 10 or 12 per cent. over that with saturated steam, with a cor- onding saving of 10 to 15 per cent. in fuel consumption.

Mechanical Stokers for Locomotives.—The use of mechanical stokers ocomotives is rapidly developing. Mr. W. S. Bartholomew[1] reports in September, 1915, there were about 1,000 such stokers in use in United States.

The special gains by the use of stokers is summed up by Mr. Bar- omew as follows:

It is evident that the railroad companies and the enginemen both it by the application of mechanical stokers to locomotives.

Locomotive capacity is increased, while, strange as it may seem, the nan's labor in shoveling coal and his suffering from the heat are erially reduced.

The railroads secure a return on their investment from the increased nage, less expensive fuel, and other economies effected, and the men, vidually, make more money per month with less effort.

Small locomotives are made larger, and large locomotives are made sible.

These and other results are being accomplished by stokers designed e applied to existing power with the necessary limitations that come eby, and much more may reasonably be expected in the future, now the stoker has established itself for permanent use, as stokers will aken seriously into account in the designing of new heavy power to uilt in the future for capacity as well as economy.

The best idea of the performance of the steam locomotive is probably ured by an examination of the conclusions of the special committee ointed to coöperate with the Pennsylvania R. R. in conducting tests

[1] See complete review of "Mechanical Stoking of Locomotives," by W. S. BAR- OMEW, *Journal* of the Franklin Institute, September, 1915.

of locomotives at the St. Louis plant during the Exposition of 1904. The conclusions are as follows:

BOILER PERFORMANCE

1. Contrary to common assumption, the results show that when forced to maximum power the large boilers delivered as much steam per unit area of heating surface as the small ones.

2. At maximum power, a majority of the boilers tested delivered 12 or more lb. of steam per square foot of heating surface per hour; two delivered more than 14 lb.; and one, the second in point of size, delivered 16.3 lb. These values expressed in terms of boiler horsepower per square foot of heating surface are 0.34, 0.40 and 0.47 respectively.

3. The two boilers holding first and second place with respect to weight of steam delivered per square foot of heating surface are those of passenger locomotives.

4. The quality of the steam delivered by the boilers of locomotives under constant conditions of operation is high, varying somewhat with different locomotives and with changes in the amount of power developed between the limits of 98.3 per cent. and 99.1 per cent.

5. The evaporative efficiency is generally maximum when the power delivered is least. Under conditions of maximum efficiency most of the boilers tested evaporated between 10 and 12 lb. of water per pound of dry coal. The efficiency falls as the state of evaporation increases. When the power developed is greatest its value commonly lies between limits of 6 and 8 lb. of water per pound of dry coal.

6. The smoke-box temperature for all boilers, when working at light power, is not far from 500°F. As the power is increased the temperature gradually rises, the maximum value depending upon the extent to which the boiler is forced. For the locomotives tested it lies between 600° and 700°F.

7. With reference to grate area, the results prove beyond question that the furnace losses due to excess air are not increased by increasing the area. In general, it appears that the boilers for which the ratio of grate surface to heating surface is largest are those of greatest capacity.

8. A brick arch in the firebox results in some increase in furnace temperature and improves the combustion of the gases.

9. The loss of heat through imperfect combustion is in most cases small, except as represented by the discharge from the stack of solid particles of fuel.

10. Relatively large firebox heating surface appears to give no advantage either with reference to capacity or efficiency. The fact seems to be that the tube-heating surface is capable of absorbing such heat as may not be taken up by the firebox.

11. The value of the Serve tube over the plain tube of the same outside diameter, either as a means for increasing capacity or efficiency, was not definitely determined.

12. The draft in the front end for any given rate of combustion, as measured in inches of water, depends upon the proportions of the locomotive and the thickness and condition of the fire. Under light power its value may not exceed an inch, but it increases rapidly as the power increases. Representative maximums derived from the tests lie between the limits of 5 in. and 8.8 in.

13. Insufficient openings in the ash-pans and the mechanism of the front end, especially the diaphragm, are shown by the tests to lead to the dissipation of considerable portions of the draft force.

THE ENGINE

14. The indicated horsepower of the modern simple freight locomotive tested may be as great as 1,000 or 1,100; that of a modern compound passenger locomotive may exceed 1,600.

15. The maximum indicated horsepower per square foot of grate surface lies, for the freight locomotives, between the limits of 31.2 and 21.1; for passenger locomotives, between 35.0 and 28.1.

16. The steam consumption per indicated horsepower-hour necessarily depends upon the conditions of speed and cut-off. For the simple freight locomotives tested the average minimum is 23.7. The consumption when developing maximum power is 23.8, and when under those conditions which proved to be least efficient, 29.0.

17. The compound locomotives tested, using saturated steam, consumed from 18.6 to 27 lb. of steam per indicated horsepower-hour. Aided by a superheater, the minimum consumption is reduced to 16.6 lb. of superheated steam per indicated horsepower-hour.

18. In general the steam consumption of simple locomotives decreases with increase of speed, while that of the compound locomotive increases. From this statement it appears that the relative advantages to be derived from the use of the compound diminish as the speed is increased.

19. Tests under a partially opened throttle show that when the degree of throttling is slight the effect is not appreciable. When the degree of throttling is more pronounced, the performance is less satisfactory than when carrying the same load with a full throttle and a shorter cut-off.

THE LOCOMOTIVE AS A WHOLE

20. The percentage of the cylinder power which appears as a stress in the drawbar diminishes with increase of speed. At 40 r.p.m. the

maximum is 94 and the minimum 77; at 280 r.p.m. the maximum is 87 and the minimum 62.

21. The loss of power between the drawbar and the cylinder is greatly affected by the character of the lubricant. It appears from the tests that the substitution of grease for oil upon axles and crankpins increases the machine friction from 75 to 100 per cent.

22. The coal consumption per dynamometer horsepower per hour for the simple freight locomotives tested is, at low speeds, not less than 3.5 lb., nor more than 4.5 lb., the value varying with the running conditions. At the highest speeds covered by the tests the coal consumption for the simple locomotives increased to more than 5 lb.

23. The coal consumption per dynamometer horsepower per hour for the compound freight locomotives tested is, for low speeds, between 2 and 3.7 lb. Results at higher speeds were obtained only from a two-cylinder compound, the efficiency of which under all conditions is shown to be very high. The coal consumption per dynamometer horsepower-hour for locomotive at the higher speed increases from 3.2 to 3.6 lb.

24. The coal consumption per dynamometer horsepower-hour for the four compound passenger locomotives tested varies from 2.2 to more than 5 lb., depending upon running conditions. In the case of all these locomotives the consumption increases rapidly as the speed is increased.

25. A comparison of the performance of the compound freight locomotives with that of the simple freight locomotives, is very favorable to the compounds. For a given amount of power the compound shows an average saving over the best simple of over 10 per cent., while the best compound shows a saving over the poorest simple of not far from 40 per cent. It should be remembered, however, that the conditions of the tests, which provide for the continuous operation of the locomotives at constant speed and load throughout the period covered by the observations, are all favorable to the compound.

26. It is a fact of more than ordinary significance that a steam locomotive is capable of developing a horsepower at the drawbar upon the consumption of but a trifle more than 2 lb. of coal per hour. This fact gives the locomotive high rank as a steam power plant.

27. It is worthy of mention that the coal consumption per horsepower-hour developed at the drawbar by the different locomotives tested presents marked differences. Some of these are easily explained from a consideration of the characteristics of the locomotives involved. Where the data are not sufficient to permit the assignment of a definite cause there can be no doubt that an extension of the study already made will reveal it.

Average Heat Balance for Test Locomotive.—

Percentages of Total Heat Available

Absorbed by the water in the boiler	52
Absorbed by the steam in the superheater	5
Lost in vaporizing moisture in the coal	5
Lost through the discharge of CO	1
Lost through the high temperature of escaping gases, the products of combustion	14
Lost through unconsumed fuel in the form of front-end cinders	3
Lost through unconsumed fuel in the form of cinders or sparks passed out of the stack	9
Lost through unconsumed fuel in the ash	4
Lost through radiation, leakage of steam and water, etc.	7
	100

Fuel Expense of Locomotives.—The fuel bills of a railroad constitute ordinarily about 10 per cent. of the total expense of operation, or from 30 to 40 per cent. of the actual cost of running the locomotive.

There were in 1906, on the railroads of the United States 51,000 locomotives. It is estimated that these locomotives consumed during the year not less than 90,000,000 tons of fuel, which is more than one-fifth of all the coal, anthracite and bituminous, mined in the country during the same period. The coal thus used cost the railroad $170,500,000.

Observations on several representative railroads have indicated that not less than 20 per cent. of the total fuel supplied to locomotives performs no function in moving trains forward. It disappears in the incidental ways just mentioned or remains in the firebox at the end of the run. The fuel consumption accounted for by the heat balance is, therefore, but 80 per cent. of the total consumed by the average locomotive in service. Applied on this basis to the total consumption of coal for the country, the heat balance may be converted into terms of tons of coal as follows:

Summary of results obtained from fuel burned in locomotives.

	Tons
1. Consumed in starting fires, in moving the locomotive to its train, in backing trains into or out of sidings, in making good safety-valve and leakage losses, and in keeping the locomotive hot while standing (estimated)	18,000,000
2. Utilized, that is, represented by the heat transmitted to water to be vaporized	41,040,000
3. Required to evaporate moisture contained by the coal	3,600,000
4. Lost through incomplete combustion of gases	720,000
5. Lost through heat of gases discharged from the stack	10,080,000
6. Lost through cinders and sparks	8,640,000
7. Lost through unconsumed fuel in the ash	2,880,000
8. Lost through radiation, leakage of steam and water, etc.	5,040,000
	90,000,000

Elimination of the Steam Locomotive.—During the past few years much has been written regarding the electrification of terminals and even of main lines. The development is slowly but surely coming. Although it is not within the field of these notes to present any lengthy discussion of this subject, yet the conclusions of the Chicago Association of Commerce presented in its report relating to "Smoke Abatement and Electrification of Railway Terminals" (1915) are of sufficient significance to warrant presentation. The conclusions regarding terminal electrification in Chicago are:

(*a*) That it is practicable from an engineering standpoint.

(*b*) That when effected it will be of economic advantage to the railroads.

(*c*) That it will present no greater element of danger to passengers and employes, if properly installed, than now exists with steam operation.

(*d*) That the most serious and difficult feature of the problem is the financial one.

"Conclusions Concerning the Feasibility of Eliminating the Steam Locomotive from the Railroad Terminals of Chicago and of Meeting all Operating Requirements without Resort to Complete Electrification." Basing judgment upon the present-day achievement,[1] the following general conclusions seem to be justified:

1. There is available at this time no form of locomotive, carrying its own power and capable of handling the traffic of the Chicago railroad terminals, which would be substituted for the steam locomotive, and there is no prospect of the immediate development of any such locomotive.

2. The design of a gasoline internal-combustion locomotive capable of handling the traffic of the Chicago terminals would involve such a multiplication of engine cylinders as to make its adoption almost, if not quite, prohibitive.

3. The adoption of a gasoline internal-combustion locomotive, should the design of such a machine become practicable, would not insure smokeless operation. As in the case of an automobile engine, such machines emit smoke when starting, and the amount of the smoke discharged is a function of the power developed.

4. The possibilities of an internal-combustion locomotive, in which the source of power is an oil engine, constitute a promising field for work. No such locomotive, possessing the power of a modern steam locomotive, has thus far been developed. The elaborate experiments of Dr. Rudolph Diesel are significant, but the results derived from them do not indicate that the problem of design has been solved.

5. The adoption of an oil-engine locomotive of the Diesel type, assuming the details of a satisfactory design to have been worked out, will

[1] April, 1914.

not in itself suffice to secure smokeless operation. Oil engines are smokeless only when the fuel and air supply are adjusted to suit the load. Whether an oil engine will be more or less objectionable, because of its smoke, than the existing steam locomotive, can be determined only by tests under service conditions.

6. The compressed-air storage locomotive, the hot-water storage locomotive and the storage-battery locomotive are all devices which, judged by the present state of the art, can be made serviceable only under special or peculiar conditions where more efficient devices cannot be used. It is not to be expected that such locomotives can be introduced for general work in the Chicago terminal.

7. There are certain short stretches of track in yards and industries to which it appears impracticable to apply any form of electric contact system; in the event of the complete elimination of the steam locomotive from the Chicago terminals, it would be practicable to work this trackage with some one of the specialized forms of locomotive described, notwithstanding the fact that no one of these locomotives is sufficient for the general work of the terminals.

8. The self-propelled motor cars of any of the various types described are most valuable for a light, diversified and not too frequent traffic. The field of usefulness for such cars within the limits of the Chicago terminals, where business is segregated and the passenger movement heavy, is not extensive.

9. The complete elimination of the steam locomotive from the railroad terminals of Chicago would, under present conditions, necessitate the abandonment of the service or the complete electrification of these terminals.

CHAPTER XX

FUELS

Solid Fuels.—The solid fuels used for power-plant purposes may be divided into the following general classes:

1. Coal.
2. Lignite.
3. Peat.
4. Wood.

According to the classification of the U. S. Geological Survey,[1] the various groups of coal and allied compounds are:

(*a*) Graphite.
(*b*) Anthracite.
(*c*) Semi-anthracite.
(*d*) Semi-bituminous.
(*e*) Bituminous.
(*f*) Sub-bituminous.
(*g*) Lignite.
(*h*) Peat.
(*i*) Wood, cellulose.

The forms of wood in common use are: wood, bagass, tan bark, straw and stubble.

Coke, charcoal and artificial fuel briquettes are fuels prepared from coal and wood.

Graphite.—Graphite cannot be burned with sufficient ease to warrant its use as a fuel. There are, however, extensive deposits of graphitic anthracite in Rhode Island and Massachusetts that have been exploited from time to time as fuel. Its composition is approximately carbon, 78 per cent.; volatile matter, 2.60 per cent.; silica, 15.06 per cent.; phosphorus, 0.045 per cent.

Under special treatment it has been made to burn in boiler furnaces, but with difficulty. Under suitable conditions it may be used for the generation of producer gas.

Anthracite.—The principal anthracite mines are in eastern Pennsylvania, although semi-anthracite coal is found in one or two other sections of the country. Anthracite is largely carbon. The commercial sizes are usually known as lump, broken, egg, stove, chestnut, pea, 1 and 2 buck-

[1] *Transactions* A.S.M.E., May, 1905.

wheat, rice and barley. These last two are known as No. 3 in the New York market. The heating value of the smaller sizes is considerably below that of the larger sizes as the amount of non-combustible earthy material is naturally higher in the smaller sizes. To handle the finer sizes to advantage for power-plant purposes requires specially constructed grates and forced draft.

Semi-bituminous Coals.—The combustible portion of semi-bituminous coals is very uniform in composition. The volatile matter is usually from 18 to 22 per cent. of the combustible matter. Such fuels are usually low in moisture, ash and sulphur, and rank among the best steaming coals in the world.

Among these coals are the Pocahontas, New River, Cumberland and Clearfield coals of Virginia, West Virginia and Maryland. These are the highest grade coals in the United States. They run low in ash, 3 to 8 per cent. and their heating value is in the neighborhood of 14,500 B.t.u. per pound or better, for the higher grades.

Bituminous Coals.—Bituminous coal is found extensively in the United States especially in Pennsylvania, Ohio, Kentucky, Tennessee, Indiana, Illinois, Iowa and Wisconsin. There is a wide variation in the ash content and heating value. In general the Eastern varieties are of a higher grade than the Western. Many bituminous coals are of the caking variety thus requiring considerable attention on the part of the firemen.

Sub-bituminous coal, as its name implies, is a grade between bituminous coal and the true lignites. It is frequently called black lignite. Some of the sub-bituminous coals, however, resemble bituminous coal so closely in physical appearance that it is hard to distinguish between them except by analysis. Its calorific value is usually less than that of bituminous coal. It is usually high in sulphur. It is found in the Rocky Mountain and Pacific States.

Lignite.—Lignite usually shows clearly a woody structure. Frequently the form of the bark of trees is plainly visible, although some lignites have more the appearance of brown clay. The lignites of the United States resemble the brown coals of Germany. The amount of moisture is very high in the lignites, often running from 30 to 40 per cent. The localities in which lignite is found are chiefly North Dakota, South Dakota, Texas, Arkansas, Louisiana, Mississippi and Alabama.

Kent states that the relation of the volatile matter and of the fixed carbon in the combustible portion of the coal enables us to judge the class to which the coal belongs, as anthracite, semi-anthracite, semi-bituminous, bituminous or lignite. Coals containing less than 7.5 per cent. volatile matter in the combustible, would be classed as anthracite, between 7.5 and 12.5 per cent. as semi-anthracite, between 12.5 and 25 per cent. as semi-bituminous, between 25 and 50 per cent. as bituminous and over 50 per cent. as lignitic coals or lignites. In the classification of the U. S. Geological Survey the sub-bituminous coals and lignites are distinguished by their structure and color rather than by analysis.

In summarizing tests of the U. S. Geological Survey (*Bulletins* 261, 290 and 323 and *Professional Paper* 48) Kent presents the following valuable table and calls attention to the fact that the table shows approximately the range of heating values per pound of combustible, as determined by the Mahler calorimeter, and the range of percentages of fixed carbon in the combustible (total of fixed carbon and volatile matter) in the coals from the several States. The extreme figures, 10,200 and 15,950 fairly represent the whole range of heating values of the combustible of the coals of the United States, but the figures for each State do not nearly cover the range of values in that State, and in some cases, as in Indiana and Illinois, the figures are much lower than the average heating values of the coals of the States.

	Fixed *C*, per cent.	B.t.u. per pound of combustible
Pennsylvania anthracite	89	14,900
West Virginia semi-bituminous	80.0 to 76.5	15,950 to 15,650
Arkansas semi-bituminous	84.0 to 77.0	15,250 to 15,500
Pennsylvania bituminous	67	15,500
West Virginia bituminous	67.5 to 55.0	15,500 to 15,000
Eastern Kentucky	60	15,000
Western Kentucky	55.0 to 50.5	14,400 to 13,700
Alabama	61.5 to 59.0	14,800 to 14,200
Kansas	62.0 to 53.5	14,800 to 14,100
Oklahoma	56.0 to 51.0	14,600 to 13,100
Missouri	50.5 to 47.0	14,300 to 12,600
Illinois	59.0 to 47.5	13,700 to 12,400
Iowa	57.0 to 53.5	13,600 to 12,700
Indiana	49	13,300
New Mexico	50.5 to 47.0	12,500 to 12,300
Wyoming	48.0 to 41.5	13,300 to 10,900
Montana	48.5	12,100
Colorado	46	11,500
North Dakota	48.5 to 42.5	10,200 to 11,400
Texas	44.5 to 34.0	10,900 to 11,000

The following analyses of representative coals of the six classes specified as given by Professor N. W. Lord are:

Class 1. Anthracite culm. Pennsylvania.
Class 2. Semi-anthracite. Arkansas.
Class 3. Semi-bituminous. West Virginia.

Class 4 (a). Bituminous coking. Connellsville, Pa.
Class 4 (b). Bituminous non-coking. Hocking Valley, Ohio.
Class 5. Sub-bituminous. Wyoming, black lignite.
Class 6. Lignite. Texas.

COMPOSITION OF ILLUSTRATIVE COALS, CAR-LOAD SAMPLES. PROXIMATE ANALYSIS OF "AIR-DRIED" SAMPLE

Class	1	2	3	4a	4b	5	6
Moisture	2.08	1.28	0.65	0.97	7.55	8.68	9.88
Vol. comb	7.27	12.82	18.80	29.09	34.03	41.31	36.17
Fixed carbon	74.32	73.69	75.92	60.85	52.57	46.49	43.65
Ash	16.33	12.21	4.63	9.09	5.85	3.52	10.30
Loss on air drying	3.40	1.10	1.10	4.20	Undet.	11.30	23.50

ULTIMATE ANALYSIS OF COAL DRIED AT 105°C.

Hydrogen	2.63	3.63	4.54	4.57	5.06	5.31	4.47
Carbon	76.86	78.32	86.47	77.10	75.82	73.31	64.84
Oxygen	2.27	2.25	2.68	6.67	10.47	15.72	16.52
Nitrogen	0.82	1.41	1.08	1.58	1.50	1.21	1.30
Sulphur	0.78	2.03	0.57	0.90	0.82	0.60	1.44
Ash	16.64	12.36	4.66	9.18	6.33	3.85	11.43

RESULTS CALCULATED TO AN ASH- AND MOISTURE-FREE BASIS

Vol. comb	8.91	14.82	19.85	32.34	39.30	47.05	45.31
Fixed carbon	91.09	85.18	80.15	67.66	60.70	52.95	54.69

ULTIMATE ANALYSIS

Hydrogen	3.16	4.14	4.76	5.03	5.41	5.50	5.05
Carbon	92.20	89.36	90.70	84.89	80.93	76.35	73.21
Oxygen	2.72	2.57	2.81	7.34	11.18	16.28	18.65
Nitrogen	0.98	1.61	1.13	1.74	1.61	1.25	1.47
Sulphur	0.94	2.32	0.60	1.00	0.87	0.62	1.62

CALORIFIC VALUE IN B.T.U. PER POUND, BY DULONG'S FORMULA

Air-dried coal	12,472	13,406	15,190	13,951	12,510	11,620	10,288
Combustible	15,286	15,496	16,037	15,511	14,446	13,235	12,889

Dulong's formula is

$$\text{total heat} = 14{,}600C + 62{,}000\left(H - \frac{0}{8}\right) + 4{,}000S$$

in which C, H, O and S represent the proportions of carbon, hydrogen, oxygen and sulphur.

An approximate formula sometimes used is

$$\text{total heat} = 154.8\ (100 - (\text{per cent. of ash} + \text{per cent. of moisture})).$$

From a table presented by Meyer ("Steam Power Plants," page 23) the following data are selected to indicate the relative value of the different classes of steam coals:

Kind of coal	Relative evaporative power	"Equivalent evaporation," pounds	Pounds of coal per square foot of grate per hour
Pocahontas, W. Va.[1]	100.0	9.5	15
Youghioheny, Pa.[2]	91.6	8.7	17
Hocking Valley, O.[2]	80.0	7.6	18
Big Muddy, Ill.[2]	80.0	7.6	20
Mt. Olive, Ill.[2]	67.5	6.4	20
Lackawanna, Pa.,[3] pea	84.0	8.0	15
Lackawanna, Pa.,[3] No. 1 buckwheat	79.0	7.5	13
Lackawanna, Pa.,[3] rice	74.0	7.0	12

[1] Semi-bituminous. [2] Bituminous. [3] Anthracite.

Peat.—Peat is defined by Davis[1] as partly decomposed and disintegrated vegetable matter that has accumulated in any place where the ordinary decay or chemical decomposition of such material has been more or less suspended, although the form and a considerable part of the structure of the plant organs are more or less destroyed.

In its natural state peat contains about 10 per cent. combustible and about 90 per cent. water.

Although peat has long been used in Europe as fuel for heating and other domestic purposes, it is but recently that it has been utilized for power purposes. Although it is estimated that in the United States, exclusive of Alaska, peat deposits cover an area of over 11,000 sq. miles, aggregating approximately 13,000,000,000 tons of available fuel, yet its use in this country can hardly be said to be beyond the experimental stage.

A good idea of the heating value of peat may be had from the following table:

[1] "The Uses of Peat" by C. A. Davis, Bureau of Mines *Bulletin* No. 16.

AIR-DRIED PEAT

Kind of peat	Locality	Water	Ash	Sulphur	Heating value, B.t.u.		
					Calories	Air-dried	Water-free
Brown, fibrous.....	Fremont, N. H......	6.34	7.93	0.69	5,161	9,290	9,920
Brown, fibrous.....	Hamburg, Mich.....	7.50	6.55	0.28	5,050	9,090	10,026
Light-brown, fibrous	Rochester, N. H.....	11.64	4.06	0.22	5,042	9,083	10,280
Dark-brown........	Westport, Conn.....	12.70	4.12	0.24	4,772	8,590	9,839
Brown, structureless	New Durham, N. H.	6.06	17.92	0.88	4,415	7,947	8,460
Brown............	New Fairfield, Conn.	9.63	7.93	0.46	4,367	7,861	8,690
Brown, fibrous.....	Westport, Conn.....	19.69	3.23	0.19	4,273	7,691	9,578
Brown............	Kent, Conn.........	12.10	7.22	0.83	4,269	7,684	8,743
Brown, fibrous.....	Cicero, N. Y........	14.57	7.42	0.25	4,209	7,576	8,869
Brown............	Black Lake, N. Y.	8.68	16.61	0.99	4,179	7,522	8,237
Brown, fibrous.....	La Martine, Wis....	9.95	16.77	0.79	4,149	7,468	8,293
Salt marsh.........	Kittery, Me.........	13.50	12.04	1.94	4,066	7,319	8,462
Black.............	Greenland, N. H.....	6.62	24.11	1.01	3,992	7,186	7,695
Light-brown, structureless..........	Waupaca, Wis.......	6.62	24.44	0.65	3,872	6,970	7,465
Brown, fibrous.....	Madison, Wis.......	8.99	18.77	0.38	3,857	6,943	7,628
Brown, sandy......	Kent, Conn.........	9.06	36.06	1.46	3,291	5,924	5,924
Black.............	N. Y........	6.52	28.50	0.57	2,867	5,161	5,521

A typical proximate analysis of a good grade of Florida peat is:

Moisture..	17.21
Volatile matter..................................	51.01
Fixed carbon....................................	24.85
Ash...	6.93
Sulphur...	0.49
B.t.u. per pound of dry fuel......................	10,082

Wood.—Wood is now little used as power-plant fuel. Dry wood consists of about 50 per cent. carbon, the remaining 50 per cent. being oxygen and hydrogen. Some woods, such as the evergreens, contain small quantities of turpentine.

The heat value of dry wood seems to run from about 6,600 B.t.u. per pound for white oak to 9,900 for long-leaf pine. The ash content varies, according to different writers, from 0.03 to 5.0 per cent.

When fresh cut the moisture content varies from 30 to 50 per cent., but after a few months of drying in the air this is reduced to 20 or 25 per cent.

Approximately 2¼ lb. of dry wood are required to equal 1 lb. of average bituminous coal. Pound for pound the fuel value of different dry woods is practically the same.

Bagass.—Bagass is the refuse cane from the sugar manufacture. The composition of bagass is approximately:

	Per cent.
Wood fiber	37 to 45
Combustible salts	10 to 9
Water	53 to 46
	100–100

and the corresponding heat value from 3,000 to 3,500 B.t.u. per pound.

E. W. Kerr reports[1] an equivalent evaporation of 2.25 lb. of water per pound of wet bagass having a net heating value of 3,256 B.t.u. per pound.

He recommends a rate of burning of not less than 100 lb. per square foot of grate per hour.

Tan Bark.—D. M. Meyers states[1] that the calorific value of spent tan averages:

9,500 B.t.u. per pound, dry.
2,665 B.t.u. per pound as fired (65 per cent. moisture).

He reports the following economic results:

Equivalent evaporation per pound of tan as fired, pounds	1.48
Equivalent evaporation per pound of dry tan, pounds	4.30

Straw.—A summary of the data given by Kent[3] is:

	Per cent. of volume	
	Dry winter-wheat straw	Mean for wheat and barley straw
C	46.2	36.0
H	5.6	5.0
N	0.4	0.5
O	43.7	38.0
Ash	4.1	4.8
Water	.....	15.7
	100.0	100.0

	B.t.u. per pound
Winter-wheat straw, dry	6,290
Winter-wheat straw, 6 per cent. water	5,770
Winter-wheat straw, 10 per cent. water	5,448
Buckwheat straw, dry	5,590
Flax straw, dry	6,750

[1] *Bulletin* 117, Louisiana Agricultural Experiment Station, Baton Rouge, La.
[2] *Transactions* A.S.M.E., vol. 31, p. 685.
[3] "Mechanical Engineers' Pocketbook," 9th edition, 1916, p. 839.

Coke.—Coke is made from coal by one of two processes. Gas-works coke is the residue from the distillation of coal in gas making. Oven coke is produced by a process of partial combustion in specially designed ovens. For fuel purposes the latter is usually the better. With the beehive-oven process of coke making the percentage yield of coal in coke averages about 60 per cent. for the United States, the range being from 44 to 75 per cent. With the byproduct coke-oven process the yield should average from 68 to 72 per cent. with good coal. The average of 29 samples of coke from six different States shows approximately the following composition:

	Per cent.
Fixed carbon	90
Ash	9
Sulphur	1

Charcoal.—By driving off the volatile matter from wood or peat by a process of partial combustion or by a process of distillation, charcoal may be produced.

The charcoal yield is from 45 to 60 bu. per cord of wood. It is far too expensive to be used much as a power-plant fuel.

Fuel Briquets.—By grinding coal, lignite or peat and pressing in forms fuel briquets may be produced. If coal is used a binder is necessary.

Investigations by the Bureau of Mines show the following binders to be commercially available:

Binder	Amount required, per cent.	Cost of binder per ton of briquets, cents
Petroleum residuum	4	45 to 60
Water-gas tar pitch	5 to 6	50 to 60
Coal-tar pitch	6.5 to 8	65 to 90

Many other binders have been used, but as a rule with less success or at greater cost. The cost of petroleum residuum is much higher now (1916) than when the report of the Bureau of Mines was made.

Lignite and peat briquets have been made with the use of no additional binder, but to date (1916) not on an extended commercial basis in the United States.

Although average values are often misleading, if properly used the following table of the average of a large number of determinations of the heating value of fuels may be of service. Unless otherwise specified the values are B.t.u. per pound of dry fuel.

	B.t.u. per pound
Anthracite coal (small)	12,500
Anthracite coal (large)	14,000
Semi-anthracite	13,400
Semi-bituminous	15,000
Bituminous coal (as fired)	12,300
Bituminous coal (dry)	13,200
Sub-bituminous	12,000
Lignite (as fired)	8,300
Lignite (dry)	11,300
Asphalt	17,000
Peat (as fired)	8,100
Peat (dry)	10,300
Wood (dry)	6,600 to 9,900
Bagass (45 to 55 per cent. water)	3,000 to 3,500
Tan bark (with 65 per cent. water)	2,700
Tan bark (dry)	9,500
Straw	5,400 to 6,700

Weight and Volume of Solid Fuels.—

One ton (2,000 lb.) of	Approximate space required, cu. ft.
Anthracite lump	28.8
Anthracite broken	30.3
Anthracite egg	30.8
Anthracite stove	31.1
Anthracite chestnut	31.9
Anthracite pea	32.8

	Max.	Avg.	Min.
Bituminous coal	45.6	37.8	34.3

	The weight of a bushel of coal, pounds
In Indiana	70
In Pennsylvania	76
In Alabama, Colorado, Georgia, Illinois, Ohio, Tennessee and West Virginia	80

Weight of a bushel of coke, pounds

Maximum	Average	Minimum
50	40	33

When buying coal it is well to remember that: 1. The heating power per pound of combustible is about constant; and more attention should be paid to the per cent. of earthy matter than to the calorific value per pound of coal.

2. The earthy matter appears to increase by about 1½ per cent. for each size of coal as it becomes smaller, but the price often diminished in a greater ratio.

3. The amount of refuse is always much in excess of the earthy matter reported by analysis.

4. With anthracite, the best qualities are indicated by the sharpest angles and the brightest appearance. If the coal is dull and shows seams and cracks, it will split up in the fire and not prove economical.

5. Bituminous coals showing whitish films or rusty stains should be avoided, as they indicate the presence of sulphur and pyrites.

The Purchase of Coal under Government and Commercial Specifications on the Basis of its Heating Value.—Until recent years, coal consumers purchased coal merely on the statement of the dealer as to its quality, relying on his integrity and on the reputation of the mine or district from which the coal was obtained. Even today this method must be followed by small consumers, by local dealers and even by some fairly large consumers. Only the Government and very large consumers, or a combination of small consumers, can offer to buy by specification at the present time.

The purchase of coal by specification is an important step toward the conservation of our national mineral resources, for it results in an increased use of the lower grades of coal. The poorer coals find a market by competing with the better grades, not as to the price per ton but as to the cost of an equal number of heat units.

Factors Affecting Value.—Some of the factors that may influence the commercial results obtained in a boiler are cost of the coal as determined by price and heating value, care in firing, design of the furnace and boiler setting, size of grate, formation of excessive amounts of clinker and ash, available draft and size of coal.

Of the physical characterisitcs of coal the following have a direct bearing on the value as a power-plant fuel:

(*a*) Moisture.
(*b*) Ash.
(*c*) Volatile matter and fixed carbon.
(*d*) Sulphur and clinker.
(*e*) Size of coal.
(*f*) Heat units.

(*a*) Moisture in the coal is worthless, costs money for freight and cartage and loss of heat.

(*b*) Non-combustible material, called ash, is worthless to the purchaser. In commercial coals this proportion of ash ranges from 4 to 25 per cent. As a rule the higher the percentage of ash, the poorer the coal. A fusible ash may be a very serious matter.

(*c*) Although furnaces designed for high-volatile coals may give results that make the coal as desirable as one low in volatile matter, yet in general the coal containing the higher percentage of fixed carbon is more efficiently handled or "burns better."

(*d*) Sulphur in the free state gives little trouble, but if combined with iron and other impurities may seriously reduce the efficiency of a furnace. Iron sulphide usually makes a fusible ash and causes clinker troubles and excessive grate-bar renewals.

(*e*) For efficient furnace results, coal should be fairly uniform in size. Very fine coals or coal dust, tend to check the draft and usually require special furnace construction. It is important to know the caking qualities of coal.

(*f*) In general the efficiency and value of coals will vary directly with the B.t.u. value, but as indicated above, the character of the ash and the form of the sulphur present, may destroy this relation. Suitable furnace construction is also an important factor.

In general, then, considerable care must be exercised in purchasing coal to meet properly the requirements imposed by local plant conditions as to character and variation of load, type of furnace, etc.

Specification Standards.—Kent[1] summarizes the standards as follows:

"*Anthracite and Semi-anthracite.*—The standard is a coal containing 5 per cent. volatile matter, not over 2 per cent. moisture, and not over 10 per cent. ash. A premium of 0.5 per cent. on the price will be given for each per cent. of volatile matter above 5 per cent. up to and including 15 per cent., and a reduction of 2 per cent. on the price will be made for each 1 per cent. of moisture and ash above the standard.

Semi-bituminous and Bituminous.—The standard is a semi-bituminous coal containing not over 20 per cent. volatile matter, 2 per cent. moisture 6 per cent. ash. A reduction of 1 per cent. in the price will be made for each 1 per cent. of volatile matter in excess of 25 per cent., and of 2 per cent. for each 1 per cent. of ash and moisture in excess of the standard.

For Western coals in which the volatile matter differs greatly in its percentage of oxygen, the above specification based on proximate analysis may not be sufficiently accurate, and it is well to introduce either the heating value as determined by a calorimeter or the percentage of oxygen. The author has proposed the following for Illinois coal:

The standard is a coal containing not over 6 per cent. moisture and 10 per cent. ash in an air-dried sample, and whose heating value is 14,500 B.t.u. per pound of combustible. For lower heating value per pound of the combustible, the price shall be reduced proportionately, and for each 1 per cent. increase in ash or moisture above the specified figures, 2 per cent. of the price shall be deducted."

The United States Government has been purchasing coal under

[1] "Mechanical Engineers' Pocketbook," 9th edition, p. 830.

specification for some years. The essential points of the Government specifications are as follows:

BITUMINOUS COAL

DESCRIPTION OF COAL DESIRED

1. The coal must be a good coal and must be adapted for successful use in the particular furnace and boiler equipment.

2. Bidders are required to specify the coal offered in terms of moisture, "as received;" ash, volatile matter, sulphur, and British thermal units, "dry coal." Such values become the standards for the coal of the successful bidder. In addition the bidders are required to give the trade name of the coal offered, the name or other designation of coal bed, name of mine or mines, location of mine or mines (town, county, and State), railroad on which mine or mines are located, and name of operator of mine or mines.

NOTE.—Bids not supplying the foregoing information may be considered informal and rejected. Coal of the description and analysis specified is herein known as the contract grade. Bidders are cautioned against specifying higher standards than can be maintained, for to do so will result in deductions in price and may result in the rejection of delivered coal or cancellation of the contract. In this connection it should be recognized that the small "mine samples" usually indicate a coal of higher economic value than that actually delivered in carload lots because of the care taken to separate extraneous matter from the coal in the "mine samples."

AWARD

3. In determining the award of this contract, consideration will be given to the quality of the coal (expressed in terms of ash in "dry coal," of moisture in coal "as received," and British thermal units in "dry coal") offered by the respective bidders, to the operating results obtained on the same and similar coals on previous contracts or by test, as well as to the price per ton.

4. Bids may be rejected from further consideration if they offer coals regarding which the Government has information that they possess unsuitable physical characteristics or excess volatile matter or sulphur or ash contents, or that they are unsatisfactory because of clinkering or excessive refuse, or because of having failed to meet the requirements of city smoke ordinances, or for other cause that would indicate that they are of a character or quality that the Government considers unsuited for use in its storage space or in its power-plant furnace equipment.

5. In order to compare bids as to the quality of the coal offered all proposals shall be adjusted to a common basis. The method used shall be to merge the four variables—ash, moisture, heating value, and price bid per ton—into one figure, the cost of 1,000,000 B.t.u., so that one bid may readily be compared with another. The procedure under this method will be as follows:

(*a*) All bids shall be reduced to a common basis with respect to moisture by dividing the price quoted in each bid by the difference between 100 per cent. and the percentage of moisture guaranteed in the bid. The adjusted bids shall be figured to the nearest tenth of a cent.

(b) The bids shall be adjusted to the same ash percentage by selecting as the standard the proposal that offers coal containing the lowest percentage of ash. The difference in ash content between any given bid and this standard shall be divided by 2 and the price in such bid, adjusted in accordance with the above, multiplied by the quotient. The result shall be added to the above adjusted price. The adjusted bids shall be figured to the nearest tenth of a cent.

(c) On the basis of the adjusted price, allowance shall then be made for the varying heat values by computing the cost of 1,000,000 B.t.u. for each coal offered. This determination shall be made by multiplying the price per ton adjusted for ash and moisture contents by 1,000,000 and dividing the result by the product of 2,000, multiplied by the number of British thermal units guaranteed.

6. After the elimination of undesirable bids the selection of the lowest bid of the remaining bids on the basis of the cost per 1,000,000 B.t.u. may be considered by the Government as a tentative award only, the Government reserving the right to have practical service test or tests made under the direction of the Bureau of Mines, the results to determine the final award of contract. The interested bidder or his authorized representative may be present at such test.

Causes for Rejection

7. It is understood that coal containing 3 per cent. more moisture, or 4 per cent. more ash, or 3 per cent. more volatile matter, or 1 per cent. more sulphur, or 4 per cent. less British thermal units than the specified guarantees as the standards for the coal hereunder contracted for, or if coal is furnished from mine or mines other than herein specified by the contractor, unless upon the written permission of the Government, shall be considered subject to rejection, and the Government may, at its option, either accept or reject the same. Should the Government have used a part of such coal subject to rejection, such shall not impair the Government's right to cause the contractor to remove the coal remaining of the delivery subject to rejection.

8. It is agreed that if the contractor furnishes coal in three consecutive deliveries, or in case more than 20 per cent. of the amount of the coal delivered to any date during the life of this contract which contains 3 per cent. more moisture, or 2 per cent. more ash, or 3 per cent. more volatile matter, or 1 per cent. more sulphur, or 2 per cent. less British thermal units than the specified guarantees as the standards for the coal hereunder contracted for, or if coal is furnished from mine or mines other than herein specified, unless upon the written permission of the Government, then this contract may, at the option of the Government, be terminated, or the Government may, at its option, purchase coal in the open market until it may become satisfied that the contractor can furnish coal equal to the standards guaranteed, and the Government shall have the right to charge against the contractor any excess in price of coal so purchased over the corrected price which would have been paid to the contractor had the coal been delivered by him.

9. The contractor shall be required to remove, without cost to the Government, within a reasonable time after notification, coal which has been rejected by the Government. Should the contractor not remove rejected coal within the said reasonable time, the Government shall then be at liberty to have the sa

coal removed from its premises and charge any and all costs incidental to its removal against the account of the contractor and to deduct the cost thereof from any money then due or thereafter to become due to the contractor.

PRICE AND PAYMENT

10. The Government hereby agrees to pay the contractor within 30 days after the completion of an order or delivery for each ton of 2,000 lb. of coal delivered and accepted in accordance with all the terms of this contract the price per ton determined by taking the analysis of the sample, or the average of the analyses of the samples if more than one sample is analyzed, collected from the coal delivered upon the basis of the price herein named adjusted as follows for variations in heating value, ash, and moisture from the standards guaranteed herein by the contractor.

(*a*) Considering the coal on a "dry-coal" basis, no adjustment in price shall be made for variations of 2 per cent. or less in the number of British thermal units from the guaranteed standard. When the variation in heat units exceeds 2 per cent. of the guaranteed standard, the adjustment shall be proportional and shall be determined by the following formula:

$$\frac{\text{B.t.u. delivered coal ("dry-coal" basis)}}{\text{B.t.u. ("dry-coal" basis) specified in contract}} \times \text{bid price} = \text{price resulting for}$$

B.t.u. variation from the standard.

The adjusted price shall be figured to the nearest tenth of a cent.

As an example, for coal delivered on a contract guaranteeing 14,000 B.t.u. on a "dry-coal" basis at a bid price of $3 per ton showing by calorific test results varying between 13,720 and 14,280 B.t.u., there would be no price adjustment. If, however, by way of further example, the delivered coal shows by calorific test 14,350 B.t.u. on a "dry-coal" basis, the price for this variation from the contract guaranty would be, by substitution in the formula:

$$\frac{14{,}350}{14{,}000} \times \$3 = \$3.075.$$

(*b*) No adjustment in price shall be made for variations of 2 per cent. or less below or above the guaranteed percentage of ash on the "dry-coal" basis. When the variation exceeds 2 per cent. the adjustment in price shall be determined as follows:

The difference between the ash content by analysis and the ash content guaranteed shall be divided by 2 and the quotient shall be multiplied by the bid price, and the result shall be added to or deducted from the British thermal units adjusted price or the bid price, if there is no British thermal unit adjustment, according to whether the ash content by analysis is below or above the percentage guaranteed. The adjustment for ash content shall be figured to the nearest tenth of a cent.

As an example of the method of determining the adjustment in cents per ton for coal containing an ash content varying by more than 2 per cent. from the standard, consider that coal for which the above-mentioned heat-unit adjustment is to be made has been delivered on a contract guaranteeing 10 per cent. ash and shows by analysis an ash content of 7.50 per cent. the adjustment in price would be d ws:

The difference between 10 and 7.50, which is 2.50, would be divided by 2, and the quotient of 1.25 multiplied by $3, resulting in an adjustment of 3.7 cts. per ton, which in this case would be an addition. The price after adjustment for the variations in heating value and ash content would be $3.075 plus $0.037, or $3.112.

(c) The price shall be further adjusted for moisture content in excess of the amount guaranteed by the contractor, the deduction being determined by multiplying the price bid by the percentage of moisture in excess of the amount guaranteed. The deduction shall be figured to the nearest tenth of a cent.

As an example, consider the coal for which the above-mentioned heat unit and ash adjustments are to be made, and as having been delivered on a contract guaranteeing 3 per cent. moisture, and that the coal shows by analysis 4.58 per cent. moisture, then the bid price would be multiplied by 1.58 (representing excess moisture), giving 4.7 cts. as the deduction per ton. The price to be paid per ton for the coal would then be $3.112, less $0.047, or $3.065.

11. If coal on visual inspection by the officer in charge appears to meet the contractor's guarantees, the said officer will have the right, immediately on the completion of an order, to make payment on 90 per cent. of the amount of the bill, based on the tonnage delivered and at the bid price per ton. The 10 per cent. withheld is to cover any deduction on account of the delivery of coal which on analysis and test is subject to an adjustment in price. If the 10 per cent. withheld should not be sufficient to cover the deduction, then the amount due the Government may be taken from any money thereafter to become due to the contractor, or may be collected from the sureties.

Sampling

12. The contractor shall have the privilege of having a representative present to witness the collection and preparation of the samples to be forwarded to the laboratory.

13. The samples shall be collected and prepared in accordance with the method given in Appendix A, attached hereto as a part of these specifications and proposals.

Analyses

14. The samples shall be immediately forwarded to the Bureau of Mines, Department of the Interior, Washington, D. C., and they shall be analyzed and tested in accordance with the method recommended by the American Chemical Society and by the use of a bomb calorimeter. The expense of such analyses and tests shall be made at no cost to the contractor. The results shall be reported by the Bureau of Mines to the officer in charge in not more than fifteen (15) days after the receipt of the sample—if more than one sample is received from the same delivery, the fifteen (15) days shall date from the receipt of the last sample taken.

Method of Sampling.—Proper sampling of coal is difficult. So much depends upon it that it must be properly done. For instructions in this field, the student is referred to the *Bulletins* of the Bureau of Mines and to the reports of Committee D-5, A. S. T. M. (1916).

Use of Briquets.—Briquets are good fuel. The only drawback is the cost of the binder, as it usually does not pay to briquet if the binder costs more than 25 cts. per ton of briquets.

An elaborate and carefully executed series of tests involving the use of natural coals and of briquets made from the same coal, previously crushed, has been carried out on a locomotive mounted at the testing plant of the Pennsylvania Railroad Co. at Altoona, Pa., under the direction of the Government Testing Station. Less extensive tests were made on several other railroads and some preliminary experiments involving the use of briquets in marine service have been made in connection with one of the Government's torpedo boats.

Results of Experiments.—The results obtained in these tests are said to sustain the following general conclusions:

1. The briquets made on the Government's machines have well withstood exposure to the weather and have suffered but little deterioration from handling.

2. In all classes of service involved by the experiments the use of briquets in place of natural coal appears to have increased the evaporative efficiency of the boilers tested.

3. The smoke produced has in no test been more dense with the briquets than with coal; on the contrary, in most tests the smoke density is said to have been less when briquets were used.

4. The use of briquets increases the facility with which an even fire over the whole area of the grate may be maintained.

5. In locomotive service the substitution of briquets for coal has resulted in a marked increase in efficiency, in an increase in boiler capacity, and in a decrease in the production of smoke. It has been specially noted that careful firing of briquets at terminals is effective in diminishing the amount of smoke produced.

General Deductions.—At the usual rate of combustion in locomotives the equivalent evaporation with either kind of briquet is 10 to 15 per cent. higher than with run-of-mine coal.

So far as blackness of smoke is concerned there seems to be little advantage in briquets over run-of-mine coal. However, the loss in sparks is less, and especially with the larger size of briquets.

It is a great deal easier to raise and to keep up steam with briquets than with run-of-mine coal as they contain no fines. Higher rates of combustion are feasible and consequently higher power, which is of especially great advantage on long grades.

As to efficiency, there is practically no difference between the two sizes of briquets, but the smaller ones are easier to handle.

Torpedo-boat Service.—In torpedo-boat service the substitution of briquets for coal improves the evaporative efficiency of the boiler. It does not appear to have affected, favorably or otherwise, the amount of smoke produced.

Steam can be raised more quickly with briquets than with run-of-mine coal.

Run-of-mine coal is transferred much more readily than briquets from the coal bunker to the fire room. With briquets the capacity of a coal bunker is reduced by 23 to 27 per cent.

Use of Low-grade Fuels.—The *Reports* of the U. S. Geological Survey show that, if the rate of increase of fuel consumption in this country that has held for the past 50 years is maintained, the supply of easily available coal will be exhausted before the middle of the next century. As is

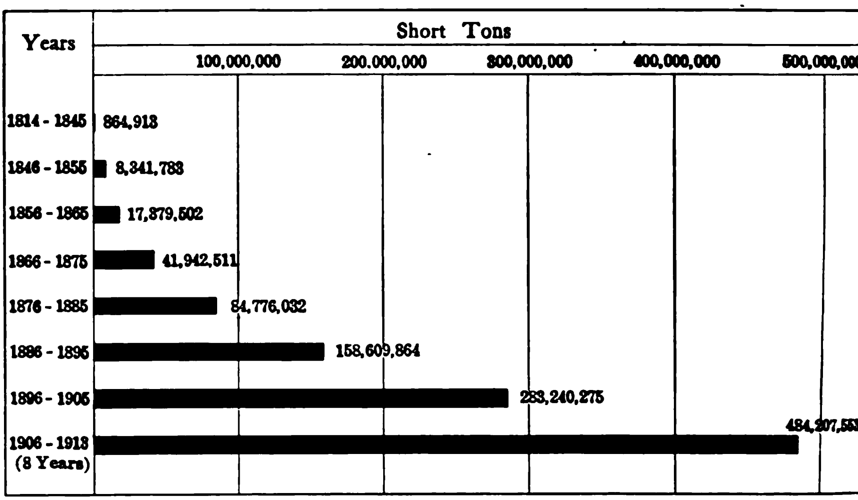


FIG. 204.—Average yearly production of coal in the United States.

shown in Fig. 204, the annual production of coal in the United States increased from less than 20,000,000 tons 60 years ago to nearly 500,000,000 tons in 1913; if the industries of the country continue to develop at a sufficient pace to maintain this rate of increase, then the limit of our coal supply will be reached in about 200 years. The fuel consumption per capita is actually increasing much faster than the population, so that the question of the continuation of this rate of increase is one of considerable importance.

It is interesting to note that the production of coal in the United States has been for some years greater than that in any other country. The world's production of coal by countries is given in Fig. 205.

Investigations into the waste of coal in mining have shown the enormous extent of this waste, aggregating from 200,000,000 to 300,000,000 tons yearly, of which at least one-half might be saved. Attention is being directed to the practicability of reducing this waste through more efficient mining methods. It has also been demonstrated that the low-

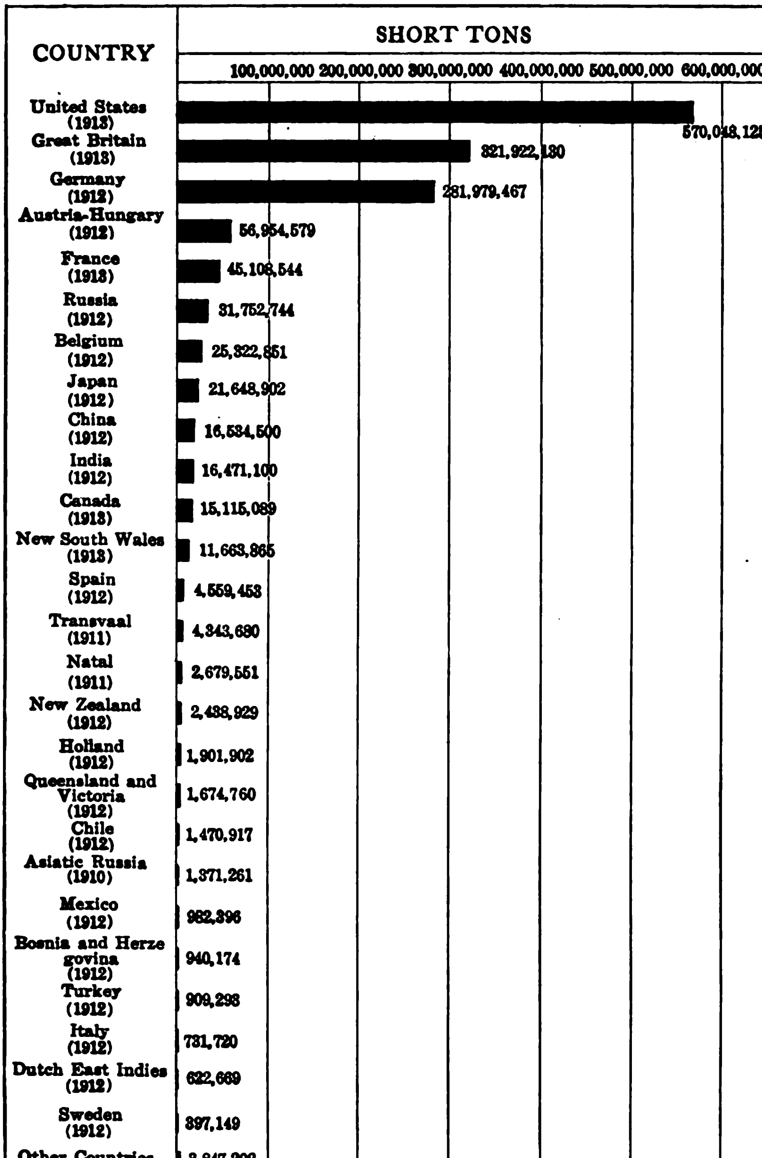


Fig. 205.—World's production of coal.

grade coals high in sulphur and ash now being left underground can be used economically in the gas producer for power and light, and should, therefore, be mined at the same time that the high-grade coal is removed.

The following low-grade fuels should, therefore, receive thoughtful consideration:

1. High-ash fuels, which are regarded at present as practically worthless.

2. Extensive deposits of lignite found in various sections of the country.

3. Peat from vast areas of swamps and bogs.

A study of the situation leads to the belief that the utilization of these fuels, which have until recently been regarded as of little or no value, may increase the fuel resources of the United States approximately (on the basis of present marketable grades):

	Per cent.
(a) Low-grade bituminous and anthracite	75 to 100
(b) Sub-bituminous, lignite and peat	60

or roughly, a total of 150 per cent.

In considering the use of such fuels, it must not be overlooked that the percentage of ash is high in the low-grade bituminous and anthracite fuels and the percentage of moisture high in many of the lignites and in the peats. These conditions practically prohibit transportation and necessitate the use of these grades in close proximity to the mine or bog.

Liquid Fuels.—The liquid fuels which are used on a large enough scale to warrant consideration as power-plant fuels are: fuel oil, gasoline, kerosene and alcohol.

The heating values and weights of these fuels run about as follows:

	Pounds per gallon	B.t.u. per pound
Fuel oil	8.3 to 6.7	18,400 to 20,400
Gasoline (high-grade)	6.0	20,500
Kerosene	6.6	19,900
Denatured alcohol	6.8	11,600

With the possible exception of denatured alcohol these fuels need no definition. Denatured alcohol as used for power purposes consists of grain alcohol (C_2H_6O) made poisonous and repulsive by the addition of wood alcohol and benzine in the following proportions: 100 parts grain alcohol, 10 parts wood alcohol, ½ part benzine.

Oil versus Coal under Boilers.—Of these fuels the only one used on a commercial basis for steam generation in boilers is fuel oil. It is, therefore, important to compare the relative results from coal and oil for this purpose.

Kent gives[1] the following table based on the assumed data: B.t.u. per pound of oil, 19,000; weight of oil, 7.57 lb. per gallon; 1 bbl. = 42 gal. = 315 lb.

[1] "Mechanical Engineers Pocketbook," 9th edition, p. 842.

Coal, B.t.u. per pound	1 lb. oil = ...pounds coal	1 bbl. oil = ...pounds coal	1 ton coal = ...barrels oil
10,000	1.9	598	3.34
11,000	1.73	544	3.68
12,000	1.58	499	4.01
13,000	1.46	460	4.34
14,000	1.36	427	4.68
15,000	1.27	399	5.01

This table shows that if coal of a heating value of only 10,000 B.t.u. per pound costs $3.34 per ton and coal of 14,000 B.t.u. per pound costs $4.68 per ton, then the price of oil will have to be as low as $1 per barrel to compete; or, on this basis oil will be the cheaper fuel if it is below $1 per barrel.

In general it may be said that the heating value of crude petroleum is from 1.44 to 1.6 times that of average good coal, even after deducting the steam used to operate the pulverizers, which steam amounts to about 4 per cent. of the total evaporation of the boilers. With the best types of apparatus this can be reduced to 2 per cent.

Under good conditions good fuel oils will evaporate from 16 to 17 lb. of water from and at 212°F. per pound of oil.

If the weight of a gallon of oil be 6.8 lb. and the cost per barrel (42 gal.) be $0.94, then the cost of 2,000 lb. would be $6.58 and at a commercial efficiency of 1 to 1.6 the values of the fuels would be the same when coal delivered, including handling of ashes, costs $4.12.

The boilers at the Chicago World's Fair gave the following average results:

Consumption of oil per hour	22.792 lb.
Equivalent evaporation per pound of oil from and at 212°	14.88 lb.
Cost of oil per hour	$56.20
Cost of oil per boiler horsepower-hour	$0.0057
Cost of labor per boiler horsepower-hour	$0.0006

Experiments on express locomotives in England gave

1 lb. of oil (max.) was equivalent to 2.4 lb. coal.
1 lb. of oil (min.) was equivalent to 2.0 lb. coal.
1 lb. of oil (avg.) was equivalent to 2.2 lb. coal.

Advantages of Liquid Fuel.—

1. Reduction in number of firemen in proportion of 5 or 6 to 1.
2. Easy lighting of fires and more regular supply of heat.
3. Fires readily regulated to suit demand for steam, and can be promptly extinguished.
4. Small proportion of refuse and its easy disposal.

5. Storage tanks can be located to best advantage, while coal bins must be near the boilers.

6. No sparks; no dust; no loss by banking.

Disadvantage of Liquid Fuel.—

1. Fire risk. Use prohibited by some city ordinances.
2. Offensive odor. Use prohibited by some cities.
3. Vapor forms explosive mixture with air.
4. Supply limited.
5. Burners make objectionable roaring noise.
6. Heating surface apt to become coated with residue.
7. Tendency of the oil to creep by valves and leak.
8. Necessity for auxiliary apparatus to start oil fire or maintain it or both.

Boiler Efficiency with Oil Fuel.—Although boiler efficiencies as high as 82 per cent. or above are reported with oil-burning furnaces, the average is probably nearer 72 per cent. if the average with coal burning furnaces be taken as 70 per cent., *i.e.*, the efficiency with oil is about 2 per cent. higher than with coal.

As the other liquid fuels are largely used in internal-combustion engines no further discussion of them will be given at this point.

Purchase of Fuel Oil under Specification.—The Bureau of Mines presents in *Technical Paper* No. 3 specifications for the purchase of fuel oil as applied by the Government. The essential features are:

1. It should not have been distilled at a temperature high enough to burn it, nor at a temperature so high that flecks of carbonaceous matter began to separate.

2. It should not flash below 60°C. (140°F.) in a closed Abel-Pensky or Pensky-Martens tester.

3. Its specific gravity should range from 0.85 to 0.96 at 15°C. (59°F.); the oil should be rejected if its specific gravity is above 0.97 at that temperature.

4. It should be mobile, free from solid or semisolid bodies, and should flow readily, at ordinary atmospheric temperatures and under a head of 1 ft. of oil, through a 4-in. pipe 10 ft. in length.

5. It should not congeal nor become too sluggish to flow at 0°C. (32°F.).

6. It should have a calorific value of not less than 10,000 calories[a] per gram (18,000 B.t.u. per pound); 10,250 calories to be the standard. A bonus is to be paid or a penalty deducted as the fuel oil delivered is above or below this standard.

7. It should be rejected if it contains more than 2 per cent. water.

[a] Calories × 1.8 = British thermal units per pound.

8. It should be rejected if it contains more than 1 per cent. sulphur.
9. It should not contain more than a trace of sand, clay or dirt.

Causes for Rejection.—1. A contract entered into under the terms of these specifications shall not be binding if, as the result of a practical service test of reasonable duration, the fuel oil fails to give satisfactory results.

2. It is understood that the fuel oil delivered during the term of the contract shall be of the quality specified. The frequent or continued failure of the contractor to deliver oil of the specified quality will be considered sufficient cause for the cancellation of the contract.

Price and Payment.—

1. Payment for deliveries will be made on the basis of the price named in the proposal for the fuel oil corrected for variations in heating value, as shown by analysis, above or below the standard fixed by the contractor. This correction is a pro rata one and the price is to be determined by the following formula:

$$\frac{\text{Delivered calories per gram (or B.t.u. per pound)} \times \text{contract price}}{\text{Standard calories per gram (or B.t.u. per pound)}} = \text{price to be paid.}$$

Water that accumulates in the receiving tank will be drawn off and measured periodically. Proper deduction will be made by subtracting the weight of the water from the weight of the oil deliveries.

Gas.—Several different kinds of gas are commercially used as fuel. The most important are:

(*a*) Natural gas.
(*b*) Illuminating gas.
(*c*) Coke-oven gas.
(*d*) Producer gas.
(*e*) Blast-furnace gas.

The heating values of these different gases vary considerably under varying conditions—the first with different geographical locations; the others with variations in the fuels used and in details of operation in their manufacture.

The following figures are fair average heating values for the gases under standard conditions (60°F. and 14.7 lb. per square inch).

		B.t.u. per cubic foot of standard gas
Natural gas		1,000
Illuminating gas		570
Coke-oven gas		550
Producer gas	Up-draft plants	150
	Double-zone plants	115
	Down-draft plants	110
Blast-furnace gas		90

Natural Gas under Steam Boilers.—Tests with natural gas under steam boilers indicate the consumption of "standard gas" per boiler horsepower to be from 38 to 60 cu. ft. in general although consumptions as high as 74 cu. ft. are reported. The corresponding efficiencies seem to range from 60 to 75 per cent. for normal commercial conditions with 1,000-B.t.u. gas. At an efficiency of 74 per cent. the consumption would be approximately 45 cu. ft. per boiler horsepower.

Absurd figures are sometimes reported which indicate test results as low as 17 or 18 cu. ft. of gas per boiler horsepower.

In addition to the figures above, it may be well to point out that even with a gas of 1,100 B.t.u. per cubic foot and a furnace efficiency of 100 per cent. the consumption must be 30.3 cu. ft. as shown below.

One boiler horsepower = 970 × 34.5 = 33,400 B.t.u. which must be transmitted to the water.

$$\frac{33,400}{1,100} = 30.3.$$

Even with this high heat value gas and an efficiency of 75 per cent. the amount of gas required per boiler horsepower will be

$$\frac{33,400}{1,100 \times 0.75} = 40.5 \text{ cu. ft.}$$

J. M. Whitham (*Transactions* A.S.M.E., 1905) gives the following conclusions as a result of a series of investigations to determine the relative value of blue flame and white flame gas under boilers:

"1. There is but little advantage possessed by one burner over another.

"2. As good economy is made with blue as with white or straw flame, and no better.

"3. Greater capacity may be made with a straw-white flame than with a blue flame.

"4. An efficiency as high as from 72 to 75 per cent. in the use of gas is seldom obtained under the most expert conditions.

"5. The 'air for dilution' is greater with gas than with coal, so possible coal efficiencies are impossible with gas.

"6. Don't expect in good commercial practice to get a boiler horsepower on less than from 43 to 45 cu. ft. of natural gas (standard).

"7. Fuel costs are the same under best conditions with natural gas at 10 cts. per 1,000 cu. ft. and semi-bituminous coal at $2.87 per 2,240 lb. (based on 3.5 lb. of wet coal per boiler horsepower-hour or 45 cu. ft. of gas).

"8. Expressed otherwise a long ton of semi-bituminous coal is equivalent to 28,700 cu. ft. of natural gas.

"9. As compared with hand-firing with coal in a plant of 1,500 boiler hp., coal being $2 per 2,240 lb., the labor saving by the use of gas is such that natural gas should sell for about 10 cts. per 1,000 cu. ft."

It has been stated that the boiler horsepower handled by one fireman is seven or eight times as much with gas-fired boilers as with coal fired.

Natural Gas for Domestic Heating.—As a result of investigations into the use of natural gas for domestic heating, W. F. M. Goss reached the following conclusions:

1. In comparison with anthracite coal, gas is worth 6.8 cts. per 1,000 cu. ft. for each $1 per ton charged for coal.

2. In comparison with bituminous coal, gas is worth 8.1 cts. per 1,000 cu. ft. for each $1 per ton charged for coal.

3. In comparison with first-class hickory wood, gas is worth 11.1 cts. per 1,000 cu. ft. for each $1 per cord charged for wood.

For example, taking values common in central Indiana, in comparison with anthracite coal at $7 per ton, gas is worth 47.6 cts.; per 1,000 cu. ft., with bituminous coal at $3.50, gas is worth 28.4 cts.; with hickory at $6 per cord, gas is worth 66.6 cts.

COST OF ENERGY IN FUELS

Kind of fuel	Cost, $	B.t.u. as fired	Number B.t.u. for $1
Small anthracite	3.00 per ton	12,500 per lb.	8,350,000
Large anthracite	7.00 per ton	14,000 per lb.	4,000,000
Bituminous coal	3.00 per ton	14,500 per lb.	9,680,000
Bituminous coal	1.50 per ton	12,300 per lb.	16,440,000
Lignite	3.00 per ton	8,300 per lb.	5,520,000
Peat	3.00 per ton	8,100 per lb.	5,400,000
Fuel oil	0.04 gal.	19,400 per lb.	3,600,000
Fuel oil	0.02 gal.	19,400 per lb.	7,300,000
Gasoline	0.30 gal.	20,500 per lb.	410,000
Gasoline	0.10 gal.	20,500 per lb.	1,230,000
Kerosene	0.30 gal.	19,900 per lb.	440,000
Kerosene	0.10 gal.	19,900 per lb.	1,320,000
Denatured alcohol	0.40 gal.	11,600 per lb.	197,000
Denatured alcohol	0.30 gal.	11,600 per lb.	263,000
Natural gas	0.30 M cu. ft.	1,000 per cu. ft.	3,333,000
Natural gas	0.10 M cu. ft.	1,000 per cu. ft.	10,000,000
Illuminating gas	0.80 M cu. ft.	570 per cu. ft.	712,000
Coke-oven gas	0.80 M cu. ft.	550 per cu. ft.	689,000
Producer gas	0.04 M cu. ft.	150 per cu. ft.	3,750,000
Producer gas	0.02 M cu. ft.	150 per cu. ft.	7,500,000
Producer gas	0.04 M cu. ft.	110 per cu. ft.	2,750,000
Producer gas	0.02 M cu. ft.	110 per cu. ft.	5,550,000
Blast-furnace gas	0.02 M cu. ft.	90 per cu. ft.	4,500,000
Blast-furnace gas	0.01 M cu. ft.	90 per cu. ft.	9,000,000

PROBLEMS

63. Check by Dulong's formula and by the approximate formula of page 376 the values of B.t.u. per pound for the fuels of page 375.

64. Does the approximate formula of page 376 give a reasonable check for the B.t.u. value for Florida peat given on page 377?

65. In response to a call for bids the following were received:

	A	B	C	D	E
B.t.u. (dry)	13,500	14,200	13,800	14,500	12,700
Moisture, per cent	1.5	2.0	3.5	1.9	6.2
Ash, per cent	8.0	9.2	7.5	5.2	8.5
Volatile matter, per cent	29.5	28.0	26.5	21.0	30.4
Sulphur, per cent	1.1	0.9	1.0	0.7	1.2
Price per ton (2,000 lb.)	$3.00	3.15	3.05	3.45	2.70

Which bid offers the lowest cost per 1,000,000 B.t.u.?

66. A coal contract specifies 13,500 B.t.u. (dry), 10 per cent. ash and 5 per cent. moisture at $2.50 per ton of 2,000 lb. delivered. The first lot of 1,000 tons averaged 13,700 B.t.u. (dry) and 12 per cent. ash and 56.2 per cent. moisture. What should be the basis of payment per ton and what is the total bonus or forfeiture for the coal company on the 1,000 tons?

67. Two boilers, one fired with oil and one with the coal delivered in problem 66, each evaporate 180,000 lb. of water as metered during a 12-hr. run. The boiler efficiency was 72 per cent. with oil and 68 per cent. with coal. With oil at 3 cts. per gallon, what was the total cost of fuel for each of the 12-hr. runs?

68. A 500-hp. boiler is run 85 per cent. above rating. Determine:

(*a*) The fuel cost for the plant for each of the fuels listed below for a period of one month of 30 days, 24 hr. per day, not including standby.

(*b*) Determine the equivalent evaporation per pound of dry coal, per pound of oil, and per 1,000 cu. ft. of gas.

Bituminous coal:

Contract	Delivered
13,500 B.t.u. dry.	14,000 B.t.u. dry.
6.5 per cent. moisture.	6.0 per cent. moisture.
8 per cent. ash in dry coal.	12.5 per cent. ash in dry coal.
$3.00 per ton (2,000 lb.).	

Oil:

19,800 B.t.u. per pound.	$0.90 per barrel (42 gal.).

Natural gas:

1,050 B.t.u. per cu. ft.	$0.25 per 1,000.

69. A 650-hp. boiler is run 200 per cent. above rating. Determine:

(*a*) The fuel cost for the plant for each of the fuels listed below for a period of 6 days, 24 hr. per day.

(*b*) The equivalent evaporation per pound of dry coal, per pound of oil, and per 1,000 cu. ft. of gas.

Bituminous coal:

Contract	Delivered
14,200 B.t.u. dry.	13,900 B.t.u. dry.
4.5 per cent. moisture.	3.8 per cent. moisture.
7.5 per cent. ash in dry coal.	6.9 per cent. ash in dry coal.
$2.20 per gross ton.	

Oil:

19,300 B.t.u. per pound.	$1.00 per barrel (42 gal.).

Natural gas:

975 B.t.u. per cubic foot.	25 cts. per 1,000.

70. (*A*) Determine the estimated fuel cost of evaporating in a steam boiler, 1,000 lb. of water, under commercial operating conditions, with each of the following fuels:

(*a*) Bituminous coal, 14,200 B.t.u. dry, 8 per cent. ash, 1.3 per cent. sulphur, 7 per cent. moisture at $3.15 per ton of 2,240 lb.
(*b*) Oil, 19,450 B.t.u. per pound at $1.20 per barrel of 42 gal., making allowance for the steam required for atomizer.
(*c*) Natural gas, 990 B.t.u. per cubic foot at 25 cts. per 1,000 cu. ft.

(*B*) With the least expensive fuel as a basis, determine the allowable cost of each of the other two fuels to make the fuel cost of evaporating 1,000 lb. of water the same for all three fuels.

(*C*) Determine the equivalent evaporation:

(*a*) Per pound of dry coal.
(*b*) Per pound of oil.
(*c*) Per 1,000 cu. ft.

CHAPTER XXI

INTERNAL-COMBUSTION ENGINES

In internal-combustion engines the pressure upon the piston is produced by the expansion of a so-called "explosive mixture." This explosive mixture consists of a combustible gas, vapor or oil mixed with air in such proportions that the mixture is easily ignited and upon ignition burns with such rapidity that a high temperature and high pressure are produced. The rate of burning is so rapid that it is commonly called an explosion.

Four-cycle and Two-cycle Engines.—Two types of internal-combustion engines are commercially in use today, the four-cycle and the two-

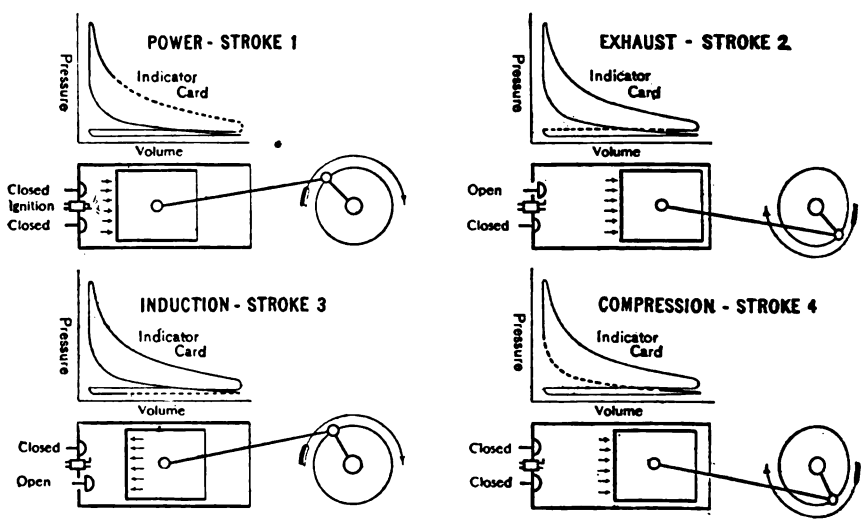


FIG. 206.

cycle. In the four-cycle, or better four-stroke cycle gas engines, the events take place as indicated by the diagram, Fig. 206.

Obviously there is, for a single-acting, single-cylinder engine but one power stroke for every two revolutions, and in commercial operation this power stroke may occasionally be missed, due to light load, improper gas mixture or failure of the ignition.

This four-stroke cycle is frequently called the "Otto-cycle" in honor of the inventor of the engine, Dr. Otto.

In the two-cycle engine the suction stroke or pump stroke of the engine and the exhaust stroke are practically done away with.

Two auxiliary pumps are used for supplying to the engine cylinder gas and air at a pressure of about 10 lb. per square inch. The exhaust valves are annular openings in the cylinder wall near the end of the cylinder and are uncovered by the piston as it nears the end of its stroke and

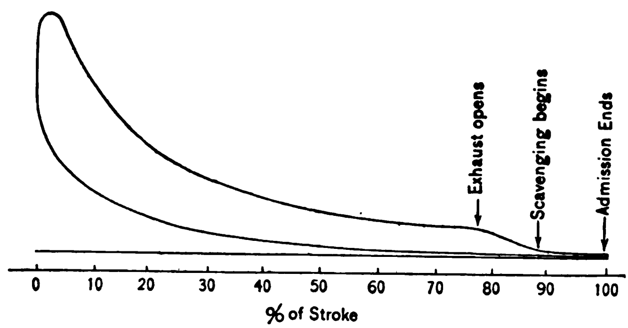


FIG. 207.—Indicator card—two-cycle gas engine.

promptly covered by the piston as it reaches the same point on the return stroke. When the piston has completed about 0.9 of its stroke the exhaust ports are uncovered, the burnt gases rush out and are followed by a rush of air from the air pump.

This air tends to "scavenge" the cylinder or free it from burnt gases. This air is immediately followed by gas in such proportions as to make an

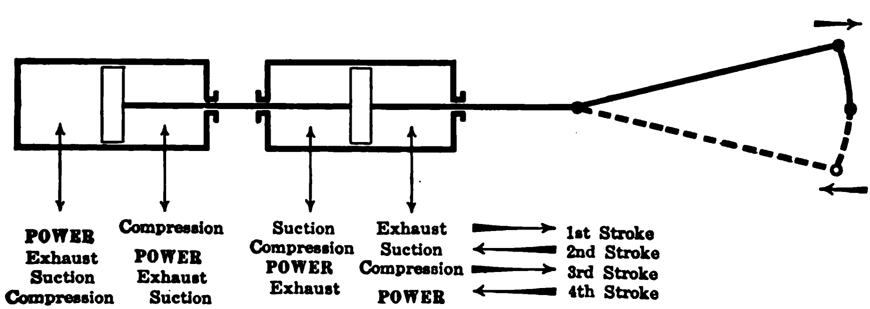


FIG. 208.—Sequence of events in the four-cycle double-acting system tandem cylinders.

explosive mixture. This mixture is compressed upon the return stroke of the piston, ignited and expanded as in the four-stroke type. The exhaust, scavenging and admission must take place in the time allowed for about 10 per cent. of one stroke. For an engine whose piston speed is 750 ft. per minute this means that these operations must be accomplished in about 0.05 sec. This short time interval means high fluid friction losses.

The irregularity of power impulses in the four-cycle engine may be readily overcome by placing two or more cylinders side by side and working all on a single crankshaft, as is widely practiced with marine steam engines. Thus the three-cylinder engine gives a power impulse regularly at every two-thirds of a revolution, which has been found sufficient for very exacting work. Cranks are spaced 120° apart, or one-third of a circumference, so that power impulses occur at successive intervals of 240° rotation.

The relative advantages and disadvantages of the two-cycle and four-cycle engines as pointed out by W. H. Adams[1] for engines of the Diesel type practically hold good for all types of two- and four-cycle internal-combustion engines. Mr. Adams states that the two-cycle type gives

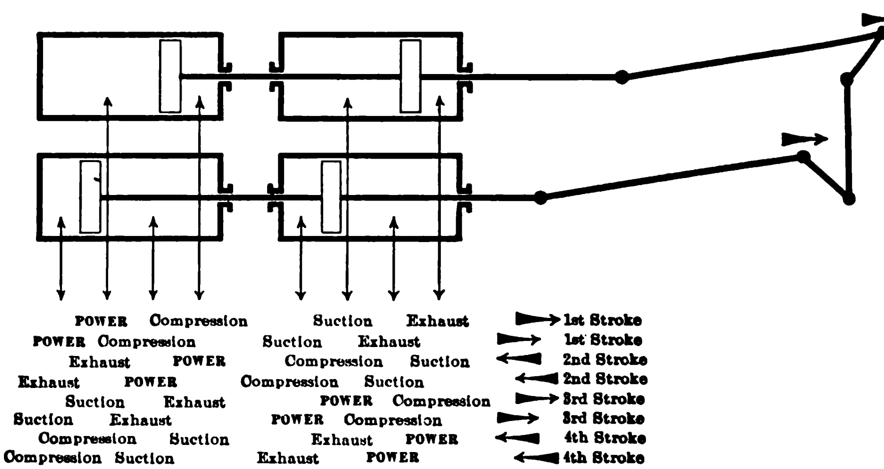


Fig. 209.—Sequence of events four-cycle, double-acting twin tandem type.

almost twice as much power for the same size of cylinder, as it has two working strokes for one in the four-cycle. (Actual value is 170 to 180 per cent.) This means less weight, less space and less first cost. As usually constructed, the piston acts as its own valve and so air inlet and exhaust valves are not required. (This is not true of some of the better class of two-cycle Diesel engines, as will be explained later.) In marine work the reduction in number of valves makes it easier to reverse a two-cycle engine. The use of the two-cycle type has also made large units possible, and single-acting engines for 1,200 hp. per cylinder have been built.

On the other hand, there is to be said for the four-cycle type of Diesel engine:

[1] "The Diesel Engine and Its Application in Southern California," by W. H. Adams, *Transactions* A.S.M.E.

(a) It is older than the two-cycle type and so has become a more stable construction.

(b) It gives better fuel economy, as expansion can be carried to the end of the stroke and no power is required for the scavenging pump. The gain is about 10 per cent.

(c) The mean temperature is lower. There is more time to remove the heat and not so much heat to remove per unit of cylinder surface. (In a two-cycle engine 90,000 B.t.u. per hour have to be removed for every square foot of cylinder surface. In four-cycle engines the figure is 40,000 B.t.u. In an ordinary water-tube boiler working at 300 per cent. of rating, it is 10,000 B.t.u.)

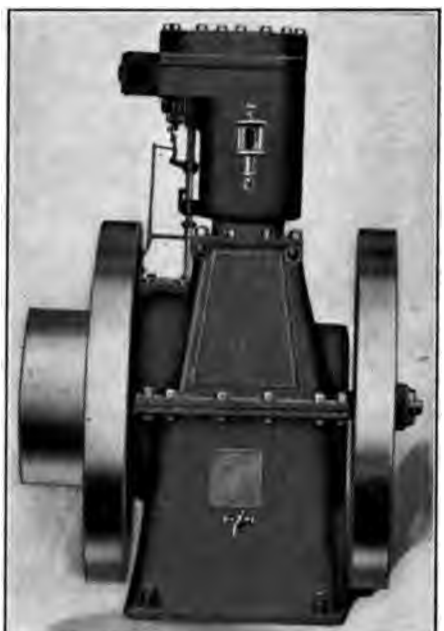


FIG. 210.—Single-cylinder, single-acting, vertical gas engine.

(d) The valve gear runs at one-half the speed of the main shaft.

(e) In the high-speed two-cycle engine, it has been difficult to get the burnt gases out of the cylinder in the short time available, so that such engines have not been quite as successful as four-cycle engines.

The tendency in this country and abroad is to use four-cycle engines up to from 700 to 1,000 hp. and above that two-cycle. This is due to the reduced first cost of the two-cycle type in the large sizes and the excessive diameter of cylinder required in large four-cycle engines. As progress is made in design, the two-cycle type may supersede the four-cycle, but this is not evident at present in the smaller sizes.

Horsepower.—The indicated horsepower of internal-combustion engines is found by the use of the same formula as for steam engines, viz:

$$\text{I.hp.} = \frac{PLAN}{33,000}$$

in which

P = m.e.p. in pounds per square inch.
L = stroke in feet.
A = effective piston area in square inches.
N = number of times per minute the pressure is exerted on the piston.

Although it is frequently convenient to determine the indicated horsepower of gas engines, and it is often desirable to do so, yet it should be remembered that although the indicated horsepower is generally used in

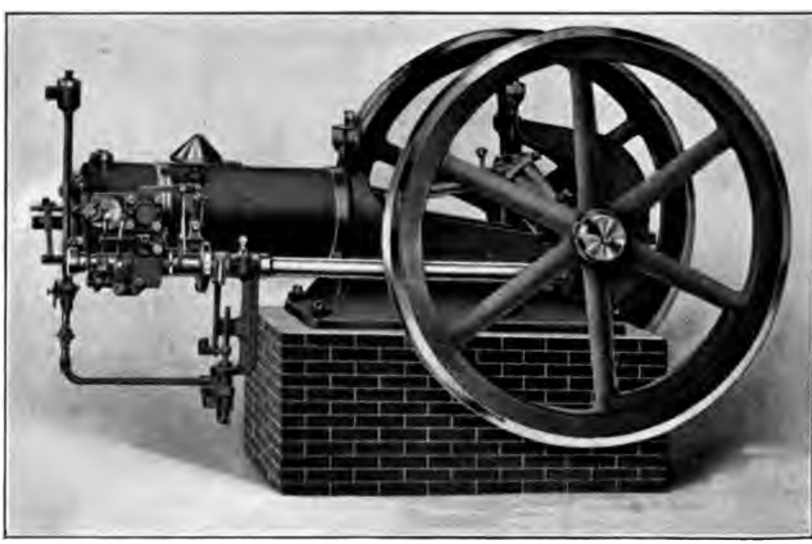


FIG. 211.—Single-cylinder, single-acting horizontal gas engine.

purchasing steam engines, the brake, or effective, horsepower is used in contracts of sale of gas engines.

In computing the brake horsepower from the cylinder dimensions and speed of four-cycle engines it is customary to assume mean effective pressures of 66, 68 or 70 lb. per square inch and a mechanical efficiency of 85 per cent.

An idea of the relation between cylinder dimensions and horsepower for two-cylinder, tandem, double-acting, four-cycle engines may be had from the following table:

Diam. cyl., in.	18	20	21	22	24	24	26	28	30	32
Stroke cyl., in.	24	24	30	30	30	36	36	36	42	42
Rev. per min.	150	150	125	125	125	115	115	115	100	100
Piston speed, ft. per min.	600	600	625	625	625	690	690	690	700	700
Rated b.hp.	260	320	370	405	490	545	630	740	855	985
Factor C.	0.8	0.8	0.84	0.84	0.85	0.95	0.93	0.94	0.95	0.96
Diam. cyl., in.	34	36	38	40	42	44	46	48	50	52
Stroke, in.	42	48	48	48	54	54	54	60	60	62
Rev. per min.	100	92	92	92	86	86	86	78	78	78
Piston speed, ft. per min.	700	736	736	736	774	774	774	780	780	780
Rated b.hp.	1,105	1,300	1,460	1,630	1,875	2,080	2,280	2,475	2,720	2,950
Factor C.	0.96	1.00	1.01	1.02	1.06	1.07	1.08	1.07	1.09	1.09

For determining the approximate horsepower of small automobile-type gasoline engines, the A.L.A.M. has adopted a formula

$$\text{b.hp.} = \frac{\text{diam.}^2 \times \text{no. cylinders}}{2.5}$$

This assumes a piston speed of 1,000 ft. per minute. On this basis the following ratings are derived, as given by Kent (page 1101, 9th edition):

Bore, in.	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0
Bore, mm.	64.0	76.0	89.0	102.0	114.0	127.0	140.0	154.0
Hp., 1 cylinder.	2.5	3.6	4.9	6.4	8.1	10.0	12.1	14.4
Hp., 2 cylinders.	5.0	7.2	9.8	12.8	16.2	20.0	24.2	28.8
Hp., 4 cylinders.	10.0	14.4	19.6	25.6	32.4	40.0	48.4	57.6
Hp., 6 cylinders.	15.0	21.6	29.4	38.4	48.6	60.0	72.6	86.4

For two-cycle engines of the power-boat type the American Power Boat Association uses:

$$\text{b.hp.} = \text{area of piston} \times \text{no. cylinders} \times \text{length of stroke} \times 1.5$$

It should be remembered that the rating of gas engines is such, due to increased economy with increase of load, that they cannot respond to heavy overload demands.

In order that the purchaser may have a definite idea of what he is buying and feel sure of an "overload leeway," the prevailing practice seems to be so to rate gas engines that they will respond to and maintain a load 20 per cent. above that specified in the contract as the normal rating of the engine.

It must not be forgotten that the power of a gas engine varies with the atmospheric pressure and consequently with change in elevation.

If p = barometric pressure at sea level,
p_e = barometric pressure at elevation,
hp. = horsepower at sea level,
hp.$_e$ = horsepower at elevation,

then

$$\text{hp.}_e = \frac{p_e}{p}\,\text{hp.}$$

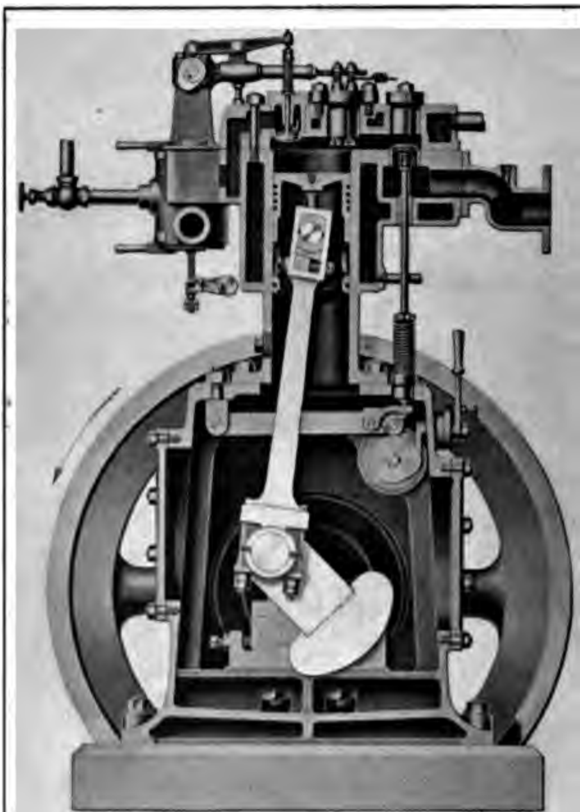


FIG. 212.—Section of single-acting, four-cycle vertical gas engine.

Piston Speeds.—Gas-engine piston speeds run approximately as lows:

Small stationary engines, 400 to 600 ft. per minute.
Large stationary engines, 500 to 1,000 ft. per minute.
Automobile engines, 600 to 1,000 ft. per minute.

Regulating or Governing.—Levin states[1] that the factors that de mine the output of an engine are: The amount of gas admitted,

[1] "Modern Gas Engine and the Gas Producer," by A. M. LEVIN, John V and Sons.

amount of air admitted, the compression effected, and the timing of the ignition.

To effect governing, two or more of these features are generally changed simultaneously.

In the hit-or-miss system the gas alone, or the gas and air, are shut off entirely at excessive speeds, but other features remain unchanged.

In throttling an already completed mixture the gas and air volumes are changed proportionally, and, thus, the quality of the charge remains unchanged, but the compression will be diminished.

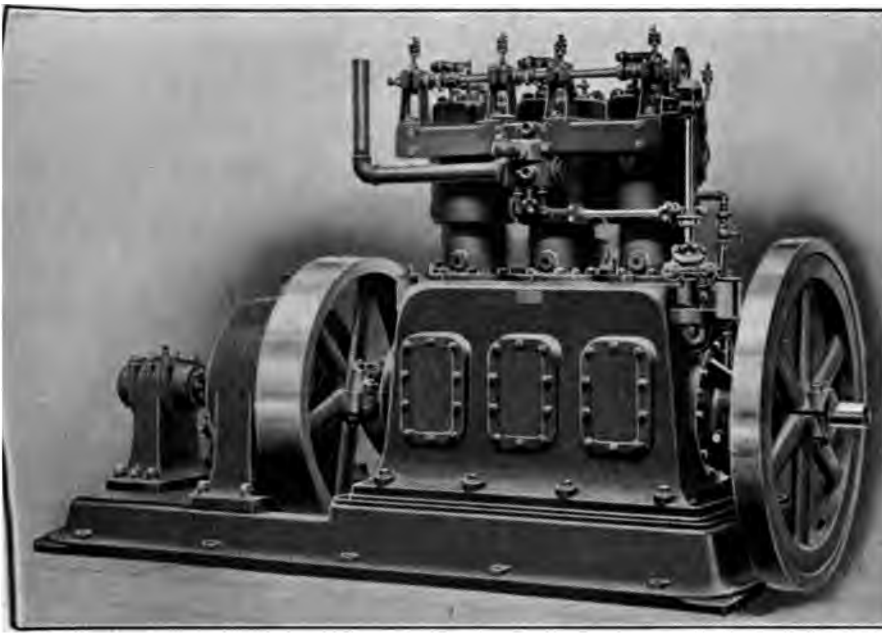


FIG. 213.—Three-cylinder, four-cycle, single-acting vertical gas engine.

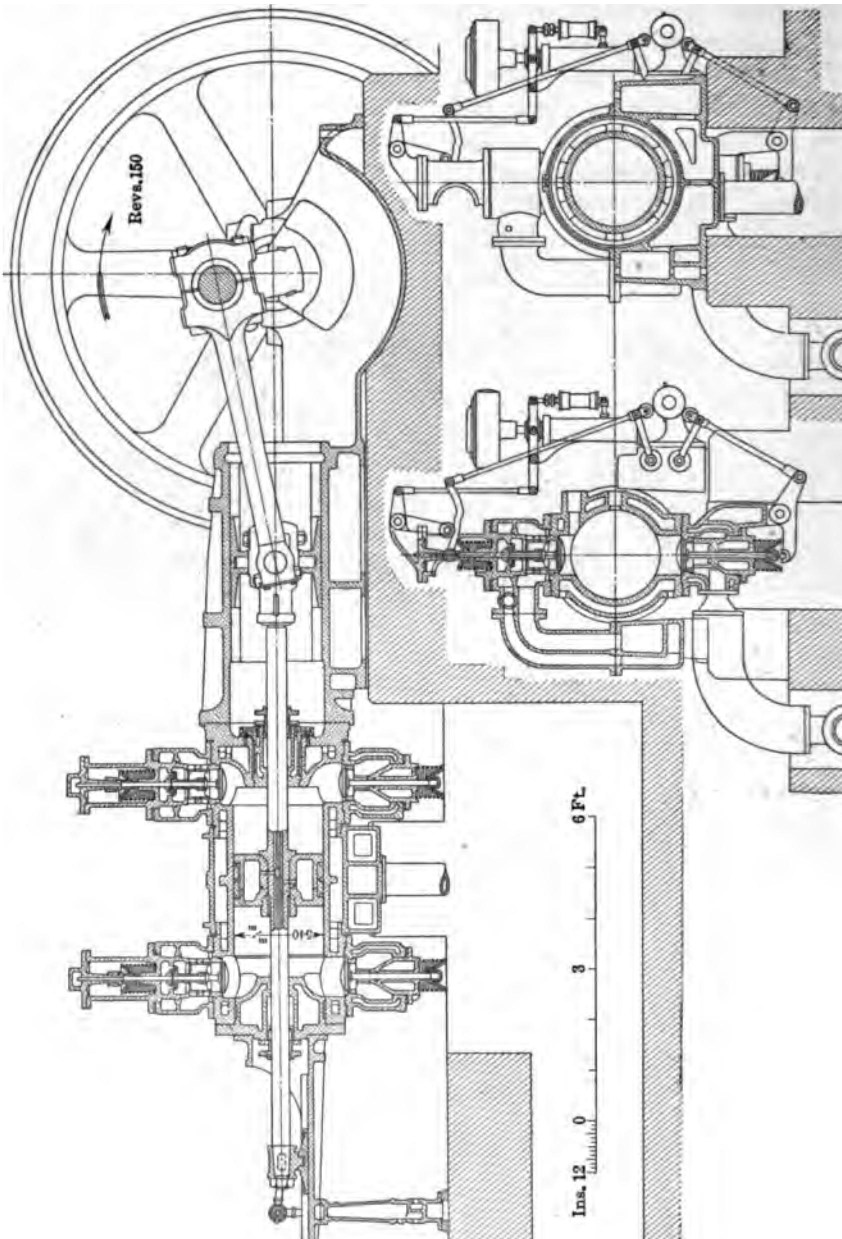
By having the gas and air throttle controlled separately, the quality of the mixture may be changed, but the quantity unchanged, and thus the compression unchanged.

Between these proportions the quality of the charge may be changed to any extent, resulting in a more or less decreased compression. It may even be possible to dilute the charge to such an extent that its quantity and compression become greater at reduced loads.

Mechanical Efficiency of Gas Engines.—Tests of both four- and two-cycle engines show the following relative mechanical efficiencies:

Four-cycle	Two-cycle
74 to 92	63 to 75
Avg. = 85	Avg. = 70.

Owing to the difficulty often encountered in obtaining the indicate horsepower of gas engines under operating conditions, and owing to th



lack of reliability in determining the indicated horsepower from indicat cards save by experienced men, it is advisable to determine the frictio horsepower of a gas engine by careful tests and thereafter use this value in determining the mechanical efficiency of the engine, as many investigations have shown the friction horsepower of such engines to be sufficiently constant to warrant this procedure.

For example, take the tests on gasoline and alcohol engines reported in U. S. Bureau of Mines, *Bulletin* No. 43.

Brake horsepower	Indicated horsepower	Friction horsepower	Per cent. rated load	Mechanical efficiency	
				$\frac{\text{b.hp.}}{\text{i.hp.}} \times 100$	$\frac{\text{b.hp.}}{\text{b.hp.} + \text{avg. f.hp.}} \times 100$
Otto 15-hp. gasoline engine					
17.16	19.34	2.18	114.3	88.8	88.3
15.98	18.30	2.37	106.7	87.3	87.6
15.40	17.70	2.30	102.8	87.0	87.1
14.80	16.90	2.10	98.6	87.5	86.7
14.18	16.22	2.04	94.5	87.4	86.2
13.66	15.87	2.21	91.0	86.1	85.8
12.41	14.70	2.29	82.9	84.4	84.6
9.98	12.29	2.31	66.5	81.4	81.5
7.60	10.18	2.58	50.7	75.3	77.0
5.09	7.49	2.40	34.0	68.0	69.2
Avg. frictional horsepower for 245 tests....		2.27			
Nash 10-hp. gasoline engine					
15.10	17.74	2.64	151.0	85.2	86.4
13.78	15.75	1.97	137.8	87.5	85.2
14.00	15.04	1.04	140.0	91.6	85.4
12.80	16.11	3.31	128.0	79.5	84.3
11.73	13.89	2.16	117.3	84.5	83.1
11.28	13.82	2.54	112.8	81.6	82.6
10.76	12.58	1.82	107.6	85.6	81.9
10.51	13.19	2.68	105.1	79.6	81.5
10.17	12.14	1.97	101.7	83.8	81.0
9.36	12.05	2.69	93.6	77.8	79.7
8.22	10.78	2.56	82.2	76.3	77.5
7.99	9.87	1.88	79.9	81.0	77.0
7.08	9.55	2.47	70.8	74.5	74.8
6.06	8.20	2.14	60.6	73.9	71.7
4.72	7.25	2.53	47.2	65.9	66.4
Avg. frictional horsepower for 104 tests....		2.39			

The figures above represent test conditions. The following data from engines operating under rather harsh conditions are, therefore, of comparative interest. These tests on pumping engines in operation in California were made under the direction of the Government.

They show that the power consumed in friction is approximately constant for a given speed, without regard to the useful work done. They also show the uneconomical results that come from using an engine too large for the work.

Brake horsepower	Indicated horsepower	Friction horsepower	Per cent. rated load	Mechanical efficiency	
				$\frac{\text{b.hp.}}{\text{i.hp.}} \times 100$	$\frac{\text{b.hp.}}{\text{b.hp.} + \text{avg. f.hp.}} \times 100$
Fairbanks-Morse 25-hp. gasoline engine					
9.3	16.9	7.6	37.2	55.0	56.4
8.0	15.2	7.2	32.0	52.6	52.6
6.7	13.4	6.7	26.8	50.0	48.2
5.4	12.6	7.2	21.6	42.8	42.8
4.0	11.6	7.6	16.0	34.5	35.7
0.0	6.7	6.7	0.0	0.0	00.0
Avg.		7.2			
White and Middleton 30-hp. gasoline engine					
11.8	18.5	6.7	39.0	64.0	64.0
8.0	16.0	7.1	26.7	50.0	54.4
5.9	11.7	5.8	19.7	50.4	46.8
2.9	10.1	7.2	9.7	28.7	30.2
Avg.		6.7			

Thermal Efficiency and Economy.—If there are no losses, 1 B.t.u. per minute would give 778 ft.-lb. per minute behind the piston, 60 B.t.u. per hour would give the same.

$$\text{B.t.u. per i.hp.-hr.} = \frac{33{,}000 \times 60}{778} = 2{,}545 \text{ with no losses of any kind.}$$

$$\text{Thermal efficiency} = \frac{\text{i.hp.} \times 33{,}000}{\text{B.t.u. per min. in fuel} \times 778}.$$

$$\text{Thermal efficiency} = \frac{\text{b.hp.} \times 33{,}000}{\text{B.t.u. per min. in fuel} \times 778}.$$

Although very wild claims are made by some manufacturers regarding the thermal efficiencies of their engines, it is probable that the maximum thermal efficiency of such engines under the most favorable operating

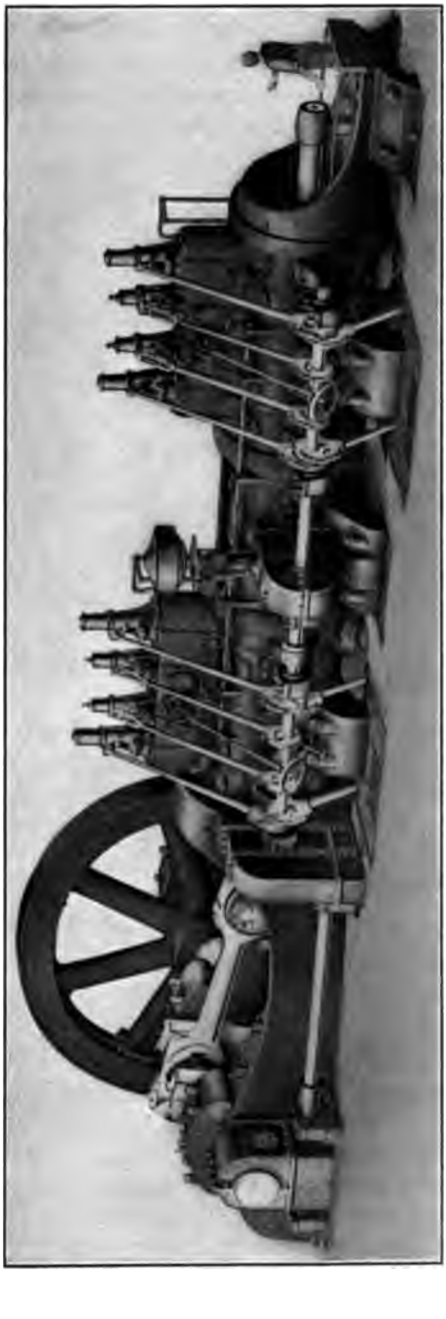
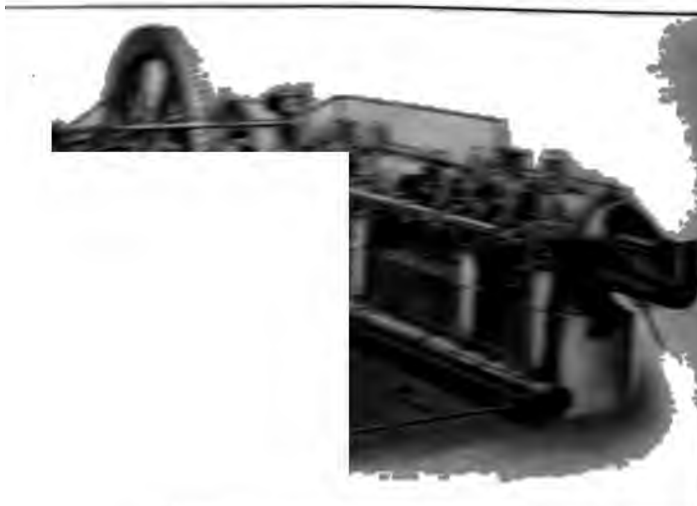


FIG. 215.—Horizontal, tandem, double-acting four-cycle gas engine.



... gas engine

... operating conditions for the same ... B.t.u. per hour.

... vary somewhat with the gas used. ... a fair basis for estimates.

Comparative Results from Denatured Alcohol and Gasoline.—The ... from denatured alcohol in internal combustion engines are ... summary of the important results from 2,000 tests ... gasoline and alcohol at the United States Bureau of Mines Testing ... presented.

The theoretically possible thermal efficiency of the Otto engine is 52 per cent. and of the Diesel 57 per cent.

	Gasoline	Alcohol
Hp. of engines	10 and 15	10 and 15
Best compression pressure, lb., sq. in	70	180
Maximum explosion pressure, lb., sq. in	....	600 to 700
Fuel per b.hp.-hr., lb	0.60	0.71
Fuel per b.hp.-hr., gal	0.10	0.10

General Conclusions.—

(*a*) For engines of the same cylinder size, but with 70 lb. compression gasoline and 180 lb. for alcohol, the maximum available horsepower the alcohol engine is about 30 per cent. greater.

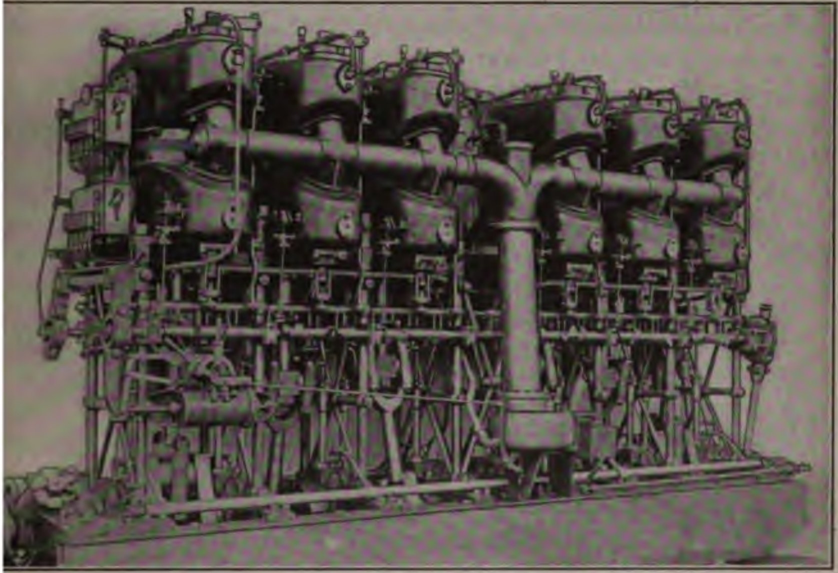


FIG. 217.—500-hp. vertical marine gas engine.

(*b*) With the compression pressures indicated, the engines required ıal volumes of gasoline and denatured alcohol, respectively, per horse-wer-hour, namely, about 1 pt.

(*c*) If alcohol be used in an engine with a compression designed for ıoline, the engine will require about 50 per cent. more alcohol than gaso-e per horsepower-hour.

(*d*) Alcohol diluted with water in any proportion up to about 50 per ıt. can be used in gasoline or alcohol engines if the engines are properly usted.

Pressures and Temperatures.—The degree of compression possible h explosive mixtures used in internal combustion engines varies with fuel used. Under normal conditions with engines working on the o cycle the allowable compression pressure will be approximately:

Fuel	Pounds per square inch
Kerosene	50 to 75
Gasoline	70 to 90
Alcohol	70 to 200
Illuminating gas	70 to 90
Natural gas	90 to 140
Producer gas	120 to 200
Blast-furnace gas	130 to 200

After combustion the pressure is much higher, usually running from 250 to 400 lb. per square inch and not infrequently reaching 600 or 700 lb. per square inch.

The temperature after combustion usually reaches 2,200°F. to 2,500°F. and may at times reach 3,000°F.

In the Diesel engine the initial compression reaches 500 to 550 lb. per square inch and the compression temperature is in the neighborhood of 1,000°F.

The two great sources of heat loss in internal combustion engines are due to the high temperature of the exhaust gases and the heat transferred through the cylinder walls to the jacket water.

The temperature of the gases at release is often from 1,500 to 1,800°F.

Circulating Water.—To remove the excess heat from the cylinder walls and in large engines from the pistons, piston rods and exhaust valves, water is circulated through cored passageways.

The amount of cooling water required per horsepower-hour is stated by different investigators as:

Investigator	Cubic feet per horsepower-hour
a	0.67 to 0.93
b	0.83 to 1.03
c	0.40 to 0.80
d	0.73 to 0.73
e	1.20 to 1.47
f	0.67 to 1.07
Average	0.75 to 1.00

The U. S. Bureau of Mines figures average for a large number of tests 0.82 cu. ft. per horsepower-hour for a three-cylinder, single-acting engine of 250-hp. rating.

The wide variation in practice in commercial plants may be seen by comparing the following figures covering long-time periods for plants in daily operation.

The high average is undoubtedly due in part to the fact that water cost little or nothing at most of these plants.

The initial temperatures reported for the cooling water for these

Plant	Cubic feet per horsepower-hour
1	3.36
2	2.80
3	2.56
4	1.01
5	2.18
6	2.56
Average	2.41

stallations range from 50° to 90°F. and the outlet temperatures from ° to 160°F., the average being 115°F.

In general there may be said to be at present a tendency toward higher mperatures of circulating water than in the past. Until within a few ears 160° was regarded as about the upper commercial limit but recent ractice in special plants has been to put the jacket water under pressure nd to increase its temperature to about 300°F. or more.

For small engines it may pay to install tanks or reservoirs for the cir-ulating water. If this is done and the circulation is maintained by the ifference in the specific gravity of the hot and cold water, the size of the anks should be sufficiently large to enable the engine to run smoothly t maximum load for several hours consecutively. The reservoirs should hen have a capacity of 50 to 65 gal. per horsepower hour.

For large installations when water is expensive, cooling towers re often installed or spray ponds built as in condensing steam-engine ractice.

Lubrication.—Owing to the high temperatures that prevail in the ylinder of the internal-combustion engine, the question of proper ubrication is a serious one. Cylinder oil should be exceedingly pure, free rom acids, and composed of hydrocarbons that leave no residue after ombustion. Only mineral oils, therefore, are suitable for the purpose. he ignition point of good cylinder oil should not be lower than 535°F. he losses in power due to poor lubrication of gas engines may amount o 10 or 15 per cent.

The amount of oil required per horsepower-hour varies with the char-cter of the installation and the method of operation. For full-load 4-hr. service, the proportion per horsepower-hour is, of course, reater than for a plant running under light load for a 9- or 10-hr. day.

The average of several figures given by the engine manufacturers for he amount of engine oil required is 0.508 gal. per 1,000 hp.-hr.

The operators of plants, however, report their commercial require-ents to be:

The average of a number of returns from the operators of reciprocating team engines indicates the consumption of cylinder oil and engine oil

GALLONS PER 1,000 HP.-HR.

Horsepower of engines	Hours of service per day	Cylinder oil	Engine oil	Other lubricants
100	8	2.0		
100	..	1.8	1.3	1.25
40, 160	16	2.8		
190	24	....	1.0	
500	24	1.25		
80, 160, 200, 375	12	1.5	3.0	
200, 500	24	....	1.26	
300	24	0.75	0.4	0.8
750	10	0.5	0.17	0.07
125	..	0.13	0.4	0.1
150, 250, 300, 600	..	0.5	1.0	
500	10	0.5	0.6	0.14
500, 1,000	10	0.4	0.6	
300, 2,000	24	2.7	5.3	0.7
115, 300, 750	24	0.25	0.5	
....	24	1.25		
Average.........	..	1.17	1.11	0.51

to be approximately the same and to equal 0.13 gal. each per 1,000 hp.-hr. On this basis the oil consumption of gas engines seems to be approximately eight or nine times as much as that of reciprocating steam engines. This is perhaps not unreasonable as the lubricating requirements of the gas engine are much more severe than those of the steam engine, but the ratio seems rather high.

Advantages of the Internal-combustion Principle.—

1. The energy of the heat liberated by combustion operates directly upon the piston of the engine to produce motion, without intervening appliances.

2. The economy in fuel per horsepower is greater than with steam. No fuel consumed wastefully in getting the motor ready to start. In plants, other than producer-gas plants, more nearly portable than steam plants.

3. Insurance lowered by absence of boiler under pressure but sometimes offset by gas-holder, or stored liquid fuel.

4. Absence of boiler avoids necessity of licensed operators.

5. Motor ready to start without previous preparation except with gas producer.

6. When fuel cut off, engine stops, not always a gain with gas producer.

7. Advantage of subdivided power, as each motor may receive its gas without loss through pipes, or from fuel tanks.

8. No storage of large amounts of energy under pressure, in a containing vessel, the rupture of which will cause disaster.

9. No boiler to cause trouble from bad water.

10. Normal and proper combustion smokeless.

11. Reduction in dust, sparks, ashes, etc., even with producers.

Disadvantages.—

1. In Otto cycle only one stroke in four is power stroke. In two-cycle only one in two. On this account for a given mean pressure a large cylinder volume is required, especially for single-acting engines.

2. Irregular crank effort. Heavy flywheel needed. If a number of cylinders are used the engine itself becomes heavy.

3. Motor does not start from rest by a simple motion of a lever or valve. This involves a clutch.

4. No way of increasing the power beyond the limit set by the diameter of the cylinder.

5. No storage of energy for overload demands, etc., as in the boiler, save in the producer system.

6. Have to cool cylinder and other parts of the engine with water.

7. Large amount of heat carried away, unutilized, by the jacket water.

8. In spite of cooling water, the valves become leaky and require attention.

9. If not carefully looked after in making the installation, the exhaust is noisy.

10. High temperature makes lubrication difficult.

11. If combustion not complete odor of exhaust offensive.

12. May get explosions in exhaust pipes or reservoirs.

13. Governing difficult on variable loads.

14. Not usually reversing in action.

15. Efficiency a maximum only near full load and when up to speed.

Rapid Development of the Gas Engine.—It was during the latter part of the nineteenth century that the gas engine found its way on to the market, and, although many types have been produced in the past 30 or 40 years, it is only within the past 10 or 15 years that the development of large engines has been noted. This development started in England, Belgium and Germany but marked progress has been limited to the past dozen years.

For many years the natural fuel of these internal-combustion engines was city gas, but even this was too expensive except for engines of small capacity. It was seldom found feasible to operate engines of more than 75 hp. on this fuel.

Cheap gas was essential for the development of the gas engine, but early attempts in this direction were somewhat discouraging, and for a time the probabilities of encroaching to any extent upon the field occupied by the steam engine were very remote.

The theoretical possibilities of the internal-combustion engine operated upon cheap fuel promised so much that the practical difficulties were rapidly overcome with the result that steam boilers and engines in many plants were replaced by gas engines, and at the present time the internal-combustion engine is a serious rival of the steam engine in many of its applications.

The development of the gas engine in point of size has been exceedingly rapid. It was in 1900 that a 600-hp. engine exhibited at the Paris Exposition was regarded as a wonder, but today four-cycle, twin-tandem, double-acting engines of 2,000 to 3,500 hp. can be found in nearly all up-to-date steel plants, and there are installations in this country containing several units rated at 5,400 hp. each.

Marine engines of the Diesel type have reached 1,200 hp. per cylinder, or 7,000 hp. in six cylinders, all single-acting.

Proper Location for a Gas Engine.—A gas engine should be located in a well-lighted place, accessible for inspection and maintenance and should be kept entirely free from dust. As a general rule the engine space should be enclosed. An engine should not be located in a cellar, on a damp floor, or in badly illuminated and ventilated places.

The pipes by which fuel is conducted to engines, the gas bags, etc., are rarely altogether free from leakage, especially if the fuel used be street gas, or natural gas. For this reason the engine room should be as well ventilated as possible in the interest of safety. Long lines of pipe between the meter and the engine should be avoided, for the sake of economy, since the chance for leakage increases with the length of pipe. Not infrequently the leakage of a pipe 30 to 50 ft. long, supplying a 30-hp. engine, may be as much as 90 cu. ft. per hour.

An engine should be supplied with gas as cool as possible, which condition is seldom realized if long pipe lines be employed for city or natural gas, extending through workshops, the temperature of which is usually higher than that of the underground piping. On the other hand, pipes should not be exposed to the freezing temperature of winter, since the frost formed within the pipe, and particularly the crystalline deposits of naphthaline, reduces the cross-section and sometimes clogs the passage. Often it happens that water condenses in the pipes; consequently, the piping should be so arranged as to avoid pockets. In places where water can collect, a drain pocket or plug should be provided so that liquid can be introduced to dissolve the naphthaline.

Starting Gas Engines.—Various methods have been used for starting these engines. Among the most common are:

1. Hand-starting with flywheel or independent crank.

2. In multi-cylinder engines, by hand pumps.
3. Compressed air [most usual today].
4. Storage of compressed explosive mixture.
5. Independent engine for starting in large plants.
6. Various explosives.

Exhaust Pipe.—If the exhaust pipe must be long, the use of elbows or sharp bends should be avoided as far as possible. In the case of very long pipes it is advisable to increase their diameter every 16 ft. from the exhaust.

For the sake of safety, at least that portion of the piping which is near the engine should be located at a proper distance from woodwork and other combustible material. Great care must be taken if the exhaust be discharged into a sewer or chimney, even though the sewer or chimney be not in use; for the unburnt gases may be trapped, and dangerous explosions may ensue at the moment of discharge.

When several engines are installed near each other, each should be provided with a special exhaust pipe, especially if the engines are to be in operation at the same time; otherwise the exhaust of one may cause excessive back-pressure on the others.

Exhaust Noises.—Among the most difficult noises to muffle is that of the exhaust. The most commonly employed means is to extend the exhaust pipe upward as far as possible, even well above the roof. This reduces the noise to some extent, but is not very efficient and produces back-pressure on the engine. Exhaust mufflers help to some extent, and the employment of pipes of sufficiently large cross-section to constitute expansion boxes in themselves will also muffle the exhaust. Considerable benefit has been derived from specially designed exhaust pipes, constructed on such lines that the gases have an opportunity for rapid expansion immediately after leaving the engine. This condition is secured by a gradual expansion of the pipe for a distance of a few feet from the engine.

A more complete solution of the problem is obtained by causing the exhaust pipe after leaving the muffler to discharge into a masonry trough having a volume equal to 12 times that of the engine cylinder. One authority states that the trough should be divided into two parts, separated by a horizontal iron grating. Into the lower part, which is empty, the exhaust pipe discharges; in the upper part paving blocks or hard stones not likely to crumble with the heat are placed. Between this layer of stones and the cover it is advisable to leave considerable space. Here the gases expand after having been divided into many parts in passing through the spaces left between adjacent stones. The trough should not be closed by a rigid cover; for although efficient muffling may be attained, yet an explosive mixture may be formed in the trough and damage caused.

The explosion is, however, less dangerous than noisy. Some authorities claim the only use of stones in the pit is to prevent the possibility of accident to careless people.

Weight of Gas Engines.—It is interesting to note the wide variation in weights per horsepower of different types of gas engines.

Type	Weight per horsepower, pounds	
Aero	2.5 to 4	
Motor boat	35 to 40	
City gas, natural gas or gasoline	200 to 250	
Oil	250 to 500	
Producer gas or blast-furnace gas	200 to 600	Avg. 300 in Europe. 400 in United States.

COST OF GAS ENGINES

Cost of Gas Engines for City or Natural Gas

Horsepower	Engine, f.o.b., dollars	Cost per horsepower, f.o.b., dollars	Horsepower	Engine, f.o.b., dollars	Cost per horsepower, f.o.b., dollars
20	700	35.00	100	3,550	35.50
20	860	43.00	100	3,830	38.30
22	775	35.20	125	4,100	32.80
25	875	35.00	125	4,475	35.80
27	1,250	46.30	135	4,200	31.10
30	1,130	37.70	140	6,980	49.90
35	1,600	45.75	150	4,856	32.40
50	1,650	33.00	160	5,230	32.70
50	1,800	36.00	175	5,750	32.80
50	1,960	39.20	175	6,275	35.80
50	2,000	40.00	195	7,300	37.40
100	3,400	34.00	200	5,600	28.00
			360	13,400	37.25

Cost of Kerosene Engines

Horsepower	Engine, f.o.b., dollars	Cost per horsepower, f.o.b., dollars	Horsepower	Engine, f.o.b., dollars	Cost per horsepower, f.o.b., dollars
1	121	121.00	10	650	65.00
2	204	102.00	15	855	57.00
4	324	81.00	20	1,060	53.00
6	444	74.00	30	1,450	48.40
8	568	71.00	40	2,020	50.50
			60	2,820	47.00

COST OF PRODUCER GAS ENGINES

orsepower	Cost, f.o.b. factory	Cost of erecting	Foundation, cubic feet	Cost of foundation	Cost of engine erected including foundation	Cost per horsepower: F.o.b. factory	Cost per horsepower: Erected, including foundation
20	1,000	...	.....	50	1,150	55.00	
55	.....	...	.....	.....	2,400	.....	43.70
60	2,800	...	350	105	......	46.70	
60	2,900	...	.....	150	......	48.40	
75	3,610	175	375	150	3,935	48.10	52.40
80	.....	...	.....	.....	3,300	.....	41.20
80	3,400	...	.....	.....	......	42.50	
80	3,250	...	.....	.....	......	40.70	
80	3,830	...	.....	.....	......	40.90	
85	4,150	...	.....	.....	......	48.90	
85	3,550	...	.....	.....	......	41.80	
100	4,925	...	.....	.....	......	49.25	
110	4,950	...	.....	225	......	45.00	
110	4,960	...	.....	.....	......	45.10	
112	4,200	...	.....	.....	......	37.50	
130	5,250	...	.....	.....	......	40.40	
135	6,600	...	.....	.....	......	48.80	
160	5,500	...	.....	.....	......	35.00	
160	6,100	150	2,000	520	6,770	38.10	42.30
250	6,650	100	2,160	560	7,360	26.60	29.40
400	12,000	...	.....	.....	......	30.00	
400	12,800	...	.....	.....	......	32.00	
600	17,400	...	.....	.....	......	29.00	
1,000	33,750	300	.....	.....	......	33.75	
2,000	64,850	875	5,400	1,400	67,125	32.43	33.56

The Oil Engine.—Although the oil engine is but a form of internal-nbustion engine and has, therefore, been reviewed in a general way in : preceding pages, it is attracting so much attention at the present ie (1916) that further details regarding it are presented.

As early as 1873 Brayton tried kerosene oil in a two-cycle engine, rning the fuel directly in the cylinder at constant pressure. Theoretic-y this should have given an efficiency of more than 50 per cent.

Unfortunately, the losses attendant on the compression of the air and : fuel, with the difficulties of finding a burner and controlling the con-nt-pressure flame became so serious that the manufacture of the engine s discontinued.

The first attempt to develop an oil engine on the Otto cycle was probably that of Priestman, who in 1888 succeeded in constructing an engine which worked on heavy petroleum distillate in a very satisfactory man-

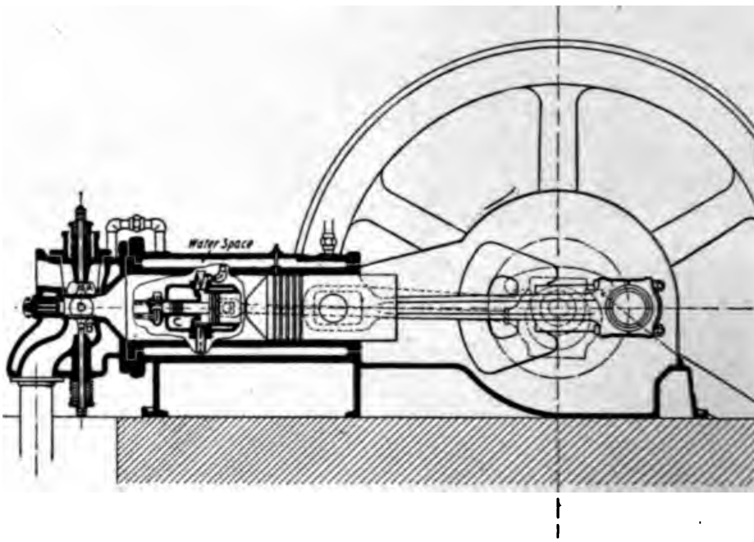


FIG. 218.—DeLaVergne type F. oil engine.

ner. Priestman used an ordinary four-cycle Otto engine, but injected his oil into a vaporizer by means of the reflex rose spray nozzle, the oil dropping into the center of the spray by gravity and the small pump

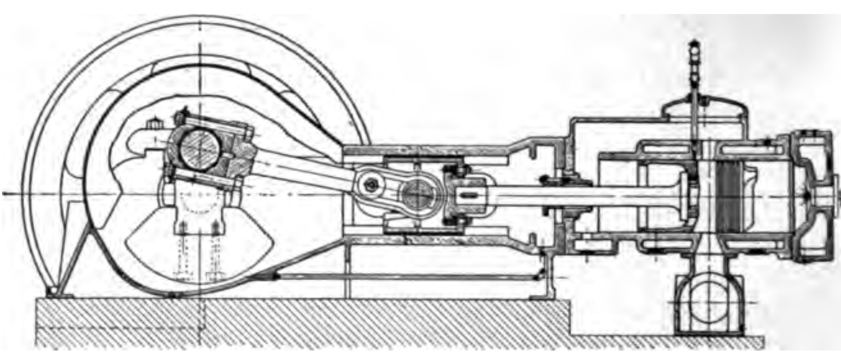


FIG. 219.—Nordberg heavy oil engine.

compressing the air for use in this apparatus. The spray was received in a cast-iron outside-heated vaporizer and the ordinary Otto cycle was carried on with this gas in the customary manner.

Most of the engines built in England from that time to this have been e or less on the Priestman principle, although the Hornsby Akro: :h came out in 1892 uses an externally heated extension of the cylinde pression space as a vaporizer, the oil being forced into this chambe neans of a small pump and as the piston returns on the compressioı ke the heating of the charge of air reaches a point at which the density ıe mixture is such that it will ignite directly from the hot chamber. :n the Brayton engine flame ignition was necessary. In the Priestman ne electrical ignition was used and with the Hornsby Akroid what unts to hot-tube ignition is the standard. Many varieties of these nes are in use today. In many of these engines there is sufficient developed in the compression space to ignite the charge, in others

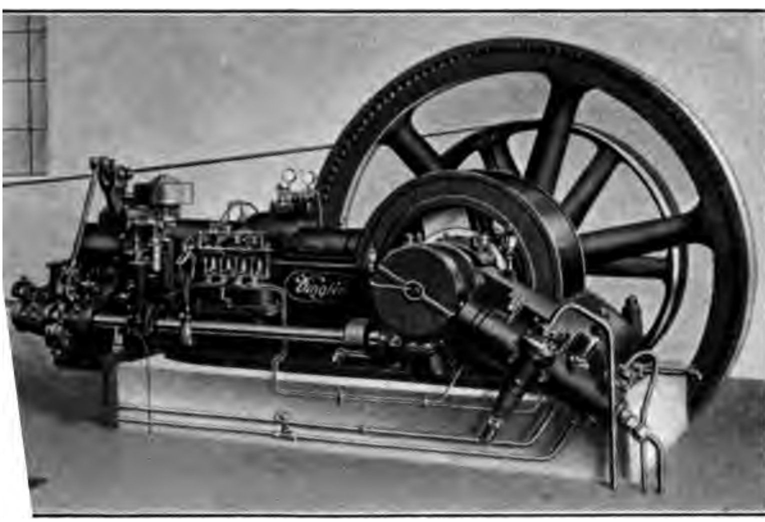


FIG. 220.—Horizontal Diesel engine, 30 b.hp.

-tube ignition must be used. Among the successful engines of ›e are the DeLaVergne, the Mietz and Weiss and many others. to be noted that most of these modern oil engines employ com-ly low compression. One hundred and fifty pounds is high, from is perhaps higher than the average. Some trouble results from ›nization of the fuel in the compression space from imperfect nd from the gumming up of the small oil passages. In most of nes the cylinder heads and the vaporizing hot tubes must be ce or twice in 24 hr. if economical running is at all a necessity. ; another type of engine in which an auxiliary compression the cylinder head is used to compress a small portion of the o the ignition point. A number of small engines have been principle.

There are three methods of securing the vaporized mixture. The first is the spray and outside-heated hot vaporizer, in which a very rich mixture of atomized oil and air is heated in a cast-iron receptacle. The entrance into the hot tube is constricted and the air admission is by a separate valve into the cylinder itself. The compression of the air on the return stroke of the piston forces a sufficient volume of air into the hot tube to get the required mixture for explosion. The second is the comparatively large vaporizer with many baffles which is heated by the exhaust gases of the engine. Into this vaporizer the oil is fed drop by drop, falling on the hot surfaces where it is vaporized. This chamber is in communication with an inlet valve and the air passes through the vaporizer making a proper mixture for explosion. The first plan is self-igniting, the second plan requires an electrical igniter.

The third system is a modification of the second in which the air supply is partly used in atomizing the fuel and is partly taken in in the ordinary way. This also requires electric ignition.

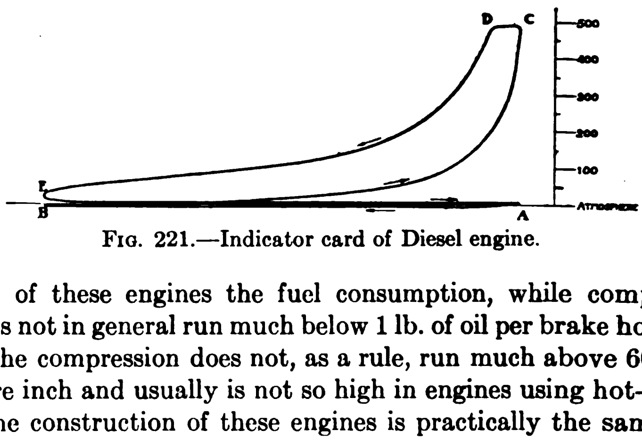


FIG. 221.—Indicator card of Diesel engine.

In all of these engines the fuel consumption, while comparatively good, does not in general run much below 1 lb. of oil per brake horsepower-hour. The compression does not, as a rule, run much above 60 or 70 lb. per square inch and usually is not so high in engines using hot-bulb ignition. The construction of these engines is practically the same as that of ordinary gas engines with the slight variations due to the vaporizer. In fact, a great many builders of engines up to 200 hp. make only slight modifications in their engines for the use of various fuels. The addition of the carburetter to the engine makes the ordinary producer gas engine fit for using gasoline. The addition of the vaporizer converts it into a kerosene engine. Otherwise the details are not modified in any way.

In 1893 Dr. Rudolf Diesel published a book entitled "The Theory and Construction of the Rational Heat Motor," in which he described a new engine with the following characteristics: First, the production of the highest temperature of the cycle not by and during combustion, but before and independently of it entirely by mechanical compression of the air. Second, the gradual introduction of a small and carefully regulated quantity of finely divided combustible into the highly compressed and heated air, in such a way that no increase of temperature takes place and all the heat generated is at once carried off by the expansion of the gases of combustion. Third, introduction of a large excess of air while maintaining a proper combustion of the fuel.

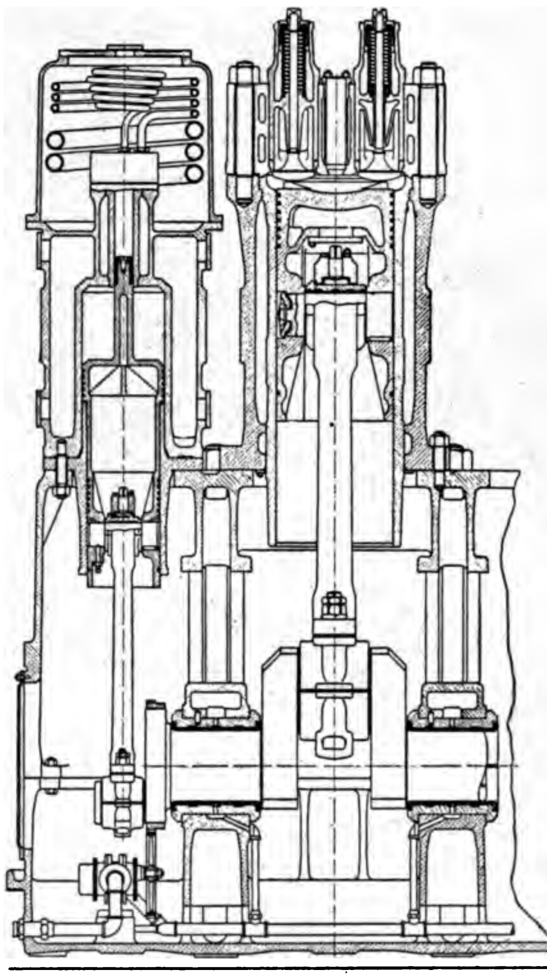


Fig. 222.—Section Busch-Sulzer Bros. Diesel engine.

This paper created great interest among engineers because of the almost revolutionary ideas which it contained. Dr. Diesel in his first proposals attempted to compress the air isothermally to pressures exceeding 250 atmospheres. This he soon found to be impossible of achievement and he modified his motor by using adiabatic compression to around 60 atmospheres. He also proposed using powdered coal as a fuel, but soon had to give this up because of the impossibility of getting rid of the ash which in a very short time stopped the working of the engine. The Diesel motor made very little progress from the date of its invention until about 1898 when small-sized engines of this type were put on the market by a number of manufacturers.

The Diesel engine at first was built on the ordinary four-cycle principle. Of late years, however, the two-cycle engine has been rather largely built, an auxiliary air pump being introduced to provide proper scavenging.

FIG. 223.—Sulzer Bros. Diesel engine, 1000 b.hp.

Although there have been many variations introduced by manufacturers, nearly all are today confining their attention to the standard Diesel principle with compression in the working cylinder up to 500 lb. per square inch (1,000°F.), using a multiple-stage air pump to provide the injection air at 600 to 850 lb. per square inch.

The extremely high pressures and temperatures of the Diesel system have put a limit to the cylinder diameter at about 30 in., which corresponds to an approximate cylinder output of say 400 hp. at 150 revolutions with four-cycle practice. It does not seem advisable to use more than six cranks on account of shafting difficulties and today the largest motors of this type might have 800 hp. per crank or 4,800 hp. for a six-cylinder engine. This power may be nearly doubled by the adoption of the two-cycle system.

The four-cycle type of engine seems to be preferable for small sizes, although difficulties with the exhaust valve are of considerable importance and increase with the size of the engine. When the two-cycle type is used, in practically all large engines, the only serious difficulties have been from the inlet valves, which usually have to be gone over about once in from 6 to 8 weeks. The horizontal type of engine may be used in small sizes, but the best results on engines of any size have been obtained with the vertical engines.

The DeLaVergne Co. in New York manufactures an engine designated as their F. H. type which operates on a variation of the Diesel principle. It is a four-cycle engine and compresses the air only to 250 to 300 lb., using a hot bulb to secure ignition.

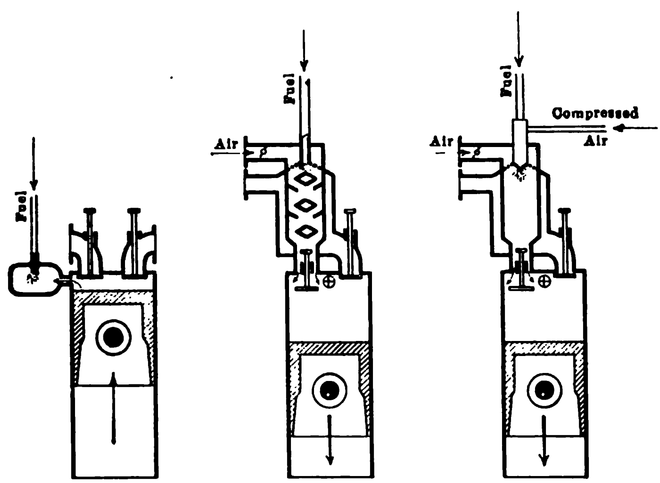


FIG. 224.—Types of oil-engine vaporizers.

There are many methods of governing a Diesel engine, but in most cases the governing is done by bypassing the oil pump so that only the proper amount of oil for the work to be done is introduced into the cylinder.

Among the auxiliaries required by a Diesel engine is an air compressor which must be of the two-stage type and have four valves. There must be an adjusting device to regulate the amount of air and it is customary to supply storage tanks, usually of the Mannesmann bottle type, in which the air is kept at a pressure of from 750 to 1,000 lb.

The regulation of the air pressure for the engine at light loads is done by hand, otherwise the large amount of air admitted with the small charge of fuel might prevent ignition.

It has been noted that the results of chemical analyses of different fuels do not furnish sufficient exact information regarding their suitability for use in the Diesel engine. This suitability can apparently only be determined by actual test. Two fuels of similar chemical analysis may give widely different results in the engine. There is a large selection of cheap fuels available, such as crude mineral oil, mineral-oil residue and gas oil, that is, the intermediate products from oil re-

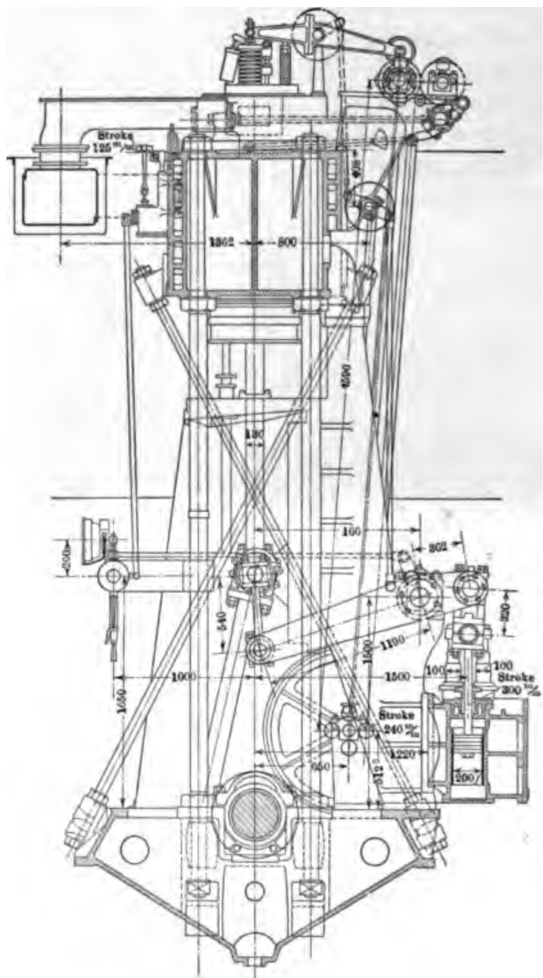


FIG. 225.—Werkspoor 1100 b.hp. marine Diesel engine.

fineries from which benzine and kerosene have been distilled, and the tar oils, tar from the water gas machine, byproducts from the distillation of coal and paraffin, wood tar and paraffin oils. When tar is used, a small amount of gasoline is first injected to insure operation before the engine is warmed up.

With such a choice of fuels it is probable that the Diesel engine will prove a favorite motor in many localities.

Piston speeds of 600 to 1,000 ft. per minute are used. Speeds lower than 200 ft. per minute are not advisable on account of the leakage and difficulties with compression.

An interesting development in the Diesel engine field is the adaptation of the Oeckelhauser type of engine to the Diesel principle. This has been done by Prof. Junkers, who has obtained 1,000 hp. from a single cylinder by his construction. In this engine three balanced cranks and connecting rods are used and the cylinders have no heads. Two pistons opposed to each other slide back and forth in the cylinder uncovering the exhaust ports at the ends of the stroke, the pumps being driven from the crossheads similarly to the ordinary marine steam engine. These engines are being built in the tandem type for marine use and promise to be of great importance, particularly for freighters whose principal business is oil carrying. The practical limit of cylinder dimensions for this type of engines will be much in excess of the 30-in. limit of the ordinary type of engine and with present materials there is little doubt that a 60-in. cylinder of 72-in. stroke could be constructed today with good results. Fullager has also built engines of this type.

At present the largest Diesel engines for land service are four-cylinder engines of approximately 2,400 b.hp. These engines are running for electric-light service with admirable results on a guaranteed oil consumption not to exceed 0.4 lb. of oil per brake horsepower-hour. They are of the two-cycle type. Four-cycle engines have an oil consumption of practically 90 per cent. of this figure, or about 0.36 lb. of oil per brake horsepower.

The thermal efficiency of these engines is between 30 and 40 per cent. Twenty-five to 30 per cent. of the heat is carried away in the cooling water and the rest in the exhaust gases.

COST OF FOUR-CYCLE DIESEL ENGINES

Horsepower	Engine, f.o.b., dollars	Cost per horsepower, f.o.b., dollars
100	7,700	77
200	12,600	63
300	17,100	57
400	21,600	54
500	26,500	53
600	30,000	50
800	39,200	49
1,000	48,000	48

About 20 per cent. of the waste heat may possibly be utilized for heating purposes and the claim is made that if a proper utilization of this heat be obtained the efficiency of the Diesel unit might be brought up to about 80 per cent.

The cost of two-cycle engines of large size is somewhat less, approximately $35 to $40 per horsepower for engines of 1,000 hp.

Foundations will cost from $2.50 to $4 per horsepower.

Erecting labor will cost from $2 to $3 per horsepower.

Summary of General Data on Diesel Engines.—

Size:

Smallest 6¾ by 8⅝, two-cycle, four-cylinder, 110 b.hp. for four cylinders.
Largest 32.2 by 39.4, two-cycle, one-cylinder, 1,250 b.hp. for one cylinder.

Weight:

250 to 500 lb. per horsepower in United States.

Speed:

150 to 250 r.p.m. Submarine service 350 to 550.
600 to 900 ft. per minute piston speed.

Mechanical Efficiency:

Per cent. rated load	30	50	75	100	120
Mechanical efficiency	43	62	70	75	78

Thermal Efficiency:

Per cent. rated load	50	75	100	120
Thermal efficiency	25	30	31	30

Economy:

Per cent. rated load	30	50	75	100	120
Pounds, oil per brake horsepower-hour (Test)	0.71	0.55	0.46	0.43	0.44
Pounds, oil per brake horsepower-hour (Mfgrs. guarantee) (Oil, 18,000 B.t.u. per pound		0.60	0.53	0.50	

Pressure for Spray:

800 to 1,100 lb. per square inch.

Air Required:

16 to 34 cu. ft. free air per brake horsepower-hour.

Power for Compressor:

4 to 7 per cent. of engine power.

Cooling Water:

0.4 to 1.2 cu. ft. per brake horsepower-hour.

Temperature Cooling Water:

130° to 140°F., max. 180°F.

Lubricating Oil:

1.25 gal. per 1,000 hp.-hr.

Attendance:

One man to 1,000 to 1,500 hp.

Life and Repairs:

Uncertain.

The Humphrey Pump.—Probably no single power-plant development has attracted more widespread attention during the past few years than the Humphrey pump. The operation of this device as described by Messrs. Potter and Trump in *Practical Engineer*, Feb. 15, 1915, is as follows:

"Operation of the Humphrey gas pump is similar to the four-stroke Otto cycle with the exception that in this pump there is complete expansion, whereas in the Otto cycle the losses from exhaust taking place under a high back-pressure are considerable.

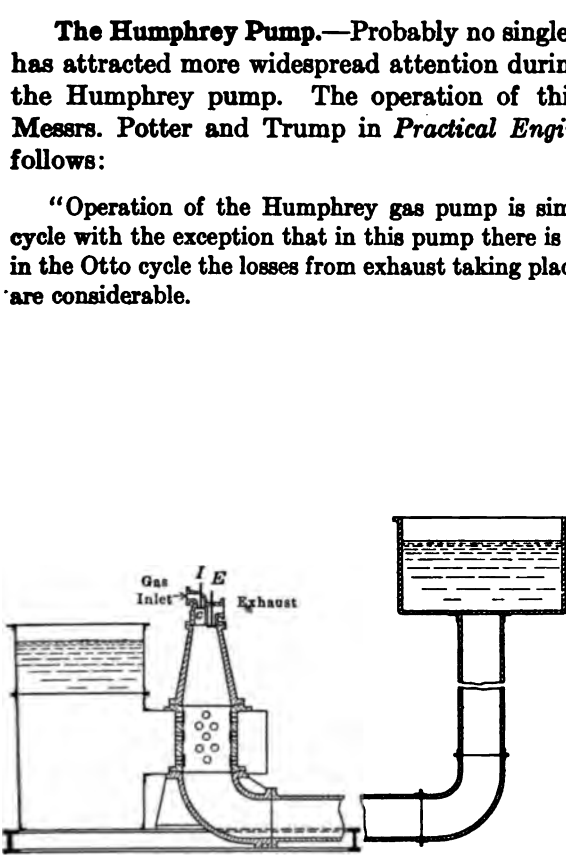


FIG. 226.—Humphrey pump.

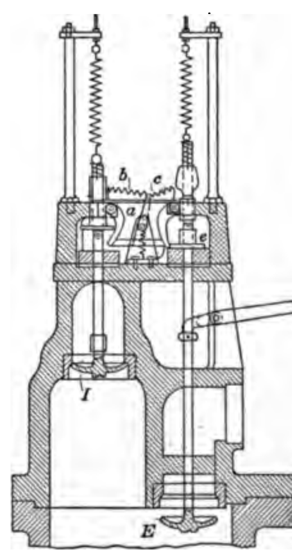


FIG. 227.—Valve gear, Humphrey pump.

"To start the pump, the proper mixture of air and gas is forced into the cylinder by a small gas-engine-driven air compressor of the two-cylinder type, one cylinder pumping air and the other gas. Two separate systems of ignition are furnished, one consisting of special spark plugs operated from storage batteries and the other from an electric generator.

"After the proper mixture of air and gas is in the cylinder, all the valves being closed, the charge is exploded by an electric spark, directly over the surface of the water, no piston or moving parts being used, and the increase in pressure resulting therefrom drives the water in the pump head downward, setting the whole column of water in the play pipe in motion. This column of water acquires kinetic energy during the period when work is being done upon it by the expanding gases. By the time these gases have expanded to atmospheric pressure, the water in the play pipe is moving at a high velocity, and as the motion of this column of water cannot be suddenly arrested, the pressure in the explosion chamber falls below atmospheric, when both scavenging and water valves open. A certain amount of water enters through the suction valves, most of which follows the moving column in the play pipe, while the rest rises in the explosion chamber. To assist the scavenging action, a certain amount of air is admitted to the explosion chamber to mix with the spent gases.

"Most of the kinetic energy in the moving column is expended in forcing water into the surge tank, and, as soon as the column of water in the play pipe comes to rest, it starts to move back toward the pump, gaining velocity until the water reaches the level of the exhaust valves, which are shut by constriction and impact. A certain quantity of the burned products mixed with the scavenging

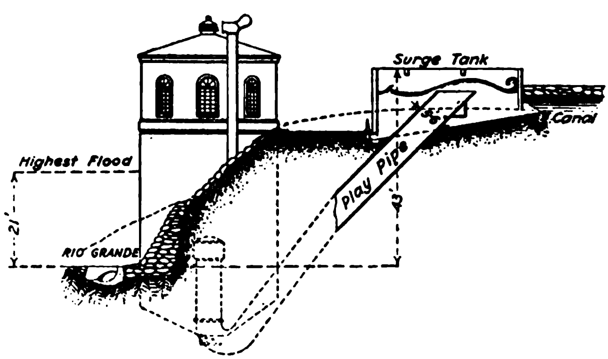


FIG. 228.—Diagram of Humphrey-pump installation.

air is now imprisoned in the cushion space and the kinetic energy of the moving column is expended in compressing this to a much greater pressure than that due to the static pumping head.

"As a result of the energy stored in these entrapped gases, the column of water is again forced outward; the pressure in the gas head is again reduced below atmospheric pressure, when a fresh charge of gas and air is drawn into the explosion chamber. Again the column of water returns and compresses the charge of gas and air which is then ignited to start a fresh cycle of operation.

"Primarily, the period of cycle of the pump is determined by the length of the reciprocating column of water in the play pipe. This motion is similar to the swing of the pendulum of a clock, and its period of vibration is governed by the length of the water column in the same way as is the period of swing of a pendulum by its length. As a general rule, assuming the column to be of uniform section, the period of vibration is almost proportional to the square root of the length of the water column."

Although extensive installations of this pump have been made in Europe and

ypt, a brief description of one of the first plants to be installed in the United
s is recorded by the same writers in the article mentioned as follows:

n reporting upon the project to irrigate certain lands in Texas along the
rande, extending from Del Rio to within 10 miles of Eagle Pass, the engi-
employed had to decide between a gravity system involving the construc-
f a supply canal some 16 miles long to water approximately 12,000 acres
d, and a pumping project to irrigate at once some 6,700 acres and to be
led to meet future needs.

Tentative plans and reports showed that the supply canal of the gravity
t could be constructed for about $300,000 with an annual expense for fixed
es, maintenance and operation of approximately $40,000 irrespective of the
ade of the canal. On the other hand, it was found the pumping project
be built for $60,000 and would entail an annual expense for fixed charges
epreciation of $6,000, and for maintenance and operation, $7,700. As the
ing project appeared so much more attractive financially than the gravity
ts, its adoption was recommended.

The pumping engine selected is made under Humphrey and Smyth patents by
umphrey Gas Pump Co., Syracuse, N. Y., and is guaranteed to pump not
han 20,000 gal. per minute against a static head of 37 ft. As it is believed
t will deliver in the neighborhood of 30,000 gal. per minute, all structures
been designed accordingly. The thermal efficiency of this pump is guaran-
y the manufacturer to be not less than 20 per cent. of heat energy in the
rned into work on the water when using producer gas having a heat value
less than 100 B.t.u. per cubic foot. The Del Rio pump will make between
d 20 complete cycles per minute."

eported test figures for an installation near London, using producer
rom anthracite, show:

Efficiency of gas plant, not including fuel used by auxiliary boiler, per cent.	82
Anthracite per water horsepower-hour pounds (guarantee was 1.1 lb.)	0.796–0.957
Thermal efficiency based on water pumped, per cent.	22–27

as Turbines.—There may be three varieties of the gas turbine: first,
ir turbine in which air is the working fluid and the furnace is outside
ystem. This turbine is analogous to the hot-air engine and may or
not have regenerative features. The air turbine is a toy and can
r be of importance, because of the impossibility of transmitting the
to the air at a sufficiently rapid rate and because of the excessive
of the pumps and other auxiliaries. The theoretical efficiency could
r exceed 10 per cent. and might be as low as 3 per cent. The com-
ial efficiency would be considerably lower. Second, the gas turbine
hich gas alone is used, predicating an inside furnace, compressors,
ierators and other complications. This turbine has more possibili-
and an efficiency of 30 per cent. might theoretically be obtained.

The size of the apparatus is large and the power used by the pumps becomes prohibitive, if an attempt at high pressures is made. High temperatures are necessary for economy and the experimental apparatus has usually burned up or fused before a test could be obtained. Third, the steam and gas turbine in which water is injected to reduce the temperature and increase the efficiency of the apparatus. The steam and gas turbine may attain an efficiency equal to the engine, or say about 35 per cent., but this efficiency is dependent very largely on the furnace temperature. At 500°F. the theoretical efficiency is 3 per cent.; at 1,000°F., 12 per cent.; at 1,500°F., 20 per cent.; and at 2,000°F. around 27 per cent. The water injection helps to carry off the heat and by regeneration these efficiencies might be somewhat increased.

There are 10 or 12 gas turbines of various kinds running at the present time. The economy, however, is not good, and in no case have real tests been reported. One of the latest machines, intended for 1,000 hp., built by Brown, Bouverie and Co. for the inventor, Holzworth, has been run somewhat successfully, but his published tests are not in a shape to quote. His machine is an air-cooled gas turbine, the air is admitted at atmospheric pressure, the gas is compressed in a centrifugal blower, while an exhaustor furnishes the vacuum. These two fans are driven by a steam turbine using steam made from the exhaust gases in a regenerator. Almost any kind of gas or oil fuel may be used, and he even used powdered Cannel coal in one of his tests. The 1,000-hp. unit weighs 25 tons and consists of a number of explosion chambers, each provided with valves, igniters and nozzle. The explosion chambers form the bedplate of the machine and the wheel is a two-stage impulse wheel with vertical shaft.

PROBLEMS

71. If a 500-hp. gas engine requires 11,500 B.t.u. per horsepower-hour at full load, how many cubic feet of each of the following gases will be required per hour when the engine is developing (A) 300 hp.; (B) 100 hp.?

(*a*) Natural gas.

(*b*) Illuminating gas.

(*c*) Up-draft producer gas.

(*d*) Down-draft producer gas.

(*e*) Blast-furnace gas.

72. Given a 100-hp. gas engine consuming 1,200 cu. ft. of natural gas per hour at full load with the barometer reading 29.35 in. and under a manometer pressure of 7 in. of water with the temperature of the gas 85°F.

The heat value of the gas is 940 B.t.u. per cubic feet as measured. Determine:

(*a*) The consumption of standard gas (60°F. and 30 in. Hg.) per horsepower-hour.

(*b*) The B.t.u. per horsepower-hour.

(*c*) The thermal efficiency (based on the heat value of the gas and the brake horsepower).

(*d*) The amount and cost of water required by this engine for one month's operation (26 days, 10 hr. per day).

73. A customer purchased a 200-hp. gas engine guaranteed to consume not over 3,300 cu. ft. of illuminating gas per hour when running at full rating.

When he received his gas bill for the first month amounting to $566 he felt that it was excessive and entered a protest. The records showed that the direct-connected D.C. generator had developed 17,800 kw.-hr. for the month, operating with uniform load 9 hr. a day for 26 days. Cost of gas $1 per 1,000 cu. ft.

1. Based on the guarantee was he justified in his protest or was the bill correct?
2. If the bill is incorrect, how much is it out?

74. Another customer with a 200-hp. gas engine guaranteed to consume not over 2,200 cu. ft. of natural gas per hour when running at full rating protested his bill of $110 for the month. The records showed that the direct-connected D.C. generator had developed 17,800 kw.-hr. for the month, operating with uniform load 9 hr. a day for 26 days. Cost of gas 30 cts. per 1,000 cu. ft.

1. Based on the guarantee was he justified in his protest or was the bill correct?
2. If the bill is incorrect, how much is it out?

75. If running under normal conditions, how many gallons of gasoline should a 250-hp. gas engine consume per 10-hr. day when developing:

(*a*) 50 hp.
(*b*) 80 hp.
(*c*) 100 hp.
(*d*) 175 hp.
(*e*) 225 hp.

76. An acceptance test of a 100-hp. gas engine operating on illuminating gas shows it to be consuming 1,810 cu. ft. of gas per hour at a load of 75 b.hp., the gas being metered at a temperature of 70°F. and under a pressure of 3 in. of water above atmosphere (barometer = 29.58). The heat value of the gas at standard conditions (60°F. and 30 in. barometer) is 600 B.t.u. per cu. ft.

The engine is guaranteed to give a full-load thermal efficiency (on b.hp.) of 25 per cent.

Would the test results justify claims of failure to meet the guarantee?

What gas consumption (as metered) should have been expected?

77. Given a 150-kw. gas power plant with direct-current generator, direct-connected to a four-cycle gas engine. Fuel, natural gas. 970 B.t.u. per cubic foot.

Determine the test economy of the plant in terms of cubic feet of gas per kilowatt-hour output at the switchboard if the demand on the plant is as follows:

6.00 a.m. to 8.30 a.m.	100 kw.
8.30 a.m. to 10.30 a.m.	150 kw.
10.30 a.m. to 3.00 p.m.	70 kw.
3.00 p.m. to 7.00 p.m.	125 kw.
7.00 p.m. to midnight	90 kw.
Midnight to 6.00 a.m.	60 kw.

78. A manufacturer is considering the installation of a generating set to deliver a rated load of 200 kw. (direct-connected). Three types of installations are under consideration:

(A) A simple, high-speed, non-condensing steam engine and direct-connected generator, hand-fired, water-tube boilers (two in service, one in reserve), closed feed-water heater, and feed pumps.

(B) A four-cycle, two-cylinder, Diesel-type oil engine with direct-connected generator, to operate on crude oil at 3 cts. per gallon.

(C) A four-cycle gas engine with generator, to operate on natural gas at 20 cts. per 1,000 cu. ft.

Coal used in the steam plant will be bituminous coal costing $3 per ton. Water costs 40 cts. per 1,000 cu. ft.

A building to house the plant is available, without cost, but foundations, stack, etc., must be provided.

The plant will carry full load 10 hr. per day, 308 days per year.

Estimate the total installation cost, the total yearly operating costs including fixed charges, and the resultant cost per kilowatt-hour generated.

CHAPTER XXII

PRODUCER GAS AND GAS PRODUCERS

Producer Gas.—Gas of some kind and quality can be made from nost anything that will burn and nearly all gases used for power, heat- ; and lighting, with the exception of natural gas, are derived from the nbustion of solid fuels or the vaporization of liquid fuels. From the ndpoint of practical convenience and economy the fuels commercially ployed for making producer gas are generally coal, coke, charcoal, lig- e and peat, although wood, sawdust, straw, oil, etc., may be advan- geously used under certain conditions.

In the most familiar process of gas making, namely the manufacture coal gas, the coal is subject to destructive distillation. The resulting s is high in illuminating qualities and has a relatively high heat value r cubic foot. In this process there is a valuable byproduct in the form coke which finds a ready market at a remunerative price.

In another process of gas making from coal a limited supply of air, h or without water vapor or steam, is passed through a thick fuel bed. the proper regulation of this air supply a partial or incomplete com- stion of the fuel is maintained resulting in the gradual consumption of entire combustible portion. Instead of having a large coke yield as yproduct, as in the former process, the coke is utilized in the gas mak- . Gas made according to this latter method is known as producer and the apparatus in which the gas is developed is called the gas ducer.

Composition of Producer Gas.—In the manufacture of any gas it ound that its definite composition will vary considerably from time time unless the details involved in the gas production are definitely ntrolled.

The essential constituents are found in all kinds of fuel gas but in h widely different proportions that the gases resulting from the differ- t systems of manufacture vary greatly in their range and manner of mmercial application.

The heat value of any gas is determined by the proportion of com- stible gases present in any mixture and by the relative percentage of ch of the individual gases. The non-combustible gases of course add thing to the heat value but rather act in the opposite direction, that is, diluents.

These combustible and non-combustible portions usually embody the following constituents in varying proportions in the different types of gas:

(1) Combustible gases
Hydrogen, H_2
Carbon monoxide, CO
Methane, CH_4 (marsh gas)
Ethylene, C_2H_4

(2) Non-combustible gases
Nitrogen, N_2
Carbon dioxide, CO_2
Oxygen, O_2

Not only is there a wide variation in the composition of various types of gases, but considerable variation will often be found in gases of the same general type.

Typical analyses of producer gas are:

COMPOSITION BY VOLUME, PER CENT.

	From anthracite	From bituminous coal	From lignite	From peat	From wood
	UP-DRAFT PLANTS				
Hydrogen, H_2	15.5	12.90	13.74	18.50	4.0
Carbon-monoxide, CO	22.7	18.28	18.72	21.00	13.6
Methane, CH_4	0.0	3.12	3.44	2.20	8.0
Ethylene, C_2H_4	0.0	0.18	0.17	0.40	0.0
Oxygen, O_2	0.3	0.04	0.16	0.00	0.0
Carbon dioxide, CO_2	5.5	9.84	10.55	12.40	12.9
Nitrogen, N_2	56.0	55.60	53.22	45.50	61.7
	100.0	100.00	100.00	100.00	100.0
	DOWN-DRAFT PLANTS				
Hydrogen, H_2	.....	12.01	14.76	10.19	
Carbon monoxide, CO	.....	21.05	16.01	16.91	
Methane, CH_4	.....	0.49	0.98	0.66	
Ethylene, C_2H_4	.....	0.01	0.00	0.06	
Oxygen, O	.....	0.13	0.01	0.41	
Carbon dioxide, CO_2	.....	6.22	11.87	10.94	
Nitrogen, N_2	.....	60.09	56.37	60.83	
		100.00	100.00	100.00	

Besides the constituents mentioned, fuel gases often contain vapors which do not appear in the analysis, but which may prove useful or detrimental in the commercial application of the gas. Many of these vapors are hydrocarbon compounds, the most familiar of which and the most important are tarry matters.

As is readily seen, the simplest producer gas is made by passing dry air through a thick bed of carbon, commercially either charcoal or coke. If a producer be filled with charcoal or coke and a fire be kindled in the lower portion then as the dry air enters from below the oxygen of the air will combine with a limited portion of the carbon in the incandescent zone thus supporting the combustion and developing a CO_2 gas. If now this carbon dioxide gas be passed through the deep bed of charcoal or coke (carbon) above the burning zone, the oxygen and carbon tend to unite to form carbon monoxide, CO, if the proper temperatures prevail. This gas is the simplest of producer gases, but it is low in heat value, and difficult to produce on an economical commercial basis. The usual procedure in producer gas making is to utilize coal as the fuel and to add a certain amount of steam with the air. This steam not only has the effect of enriching the gas but also tends to reduce the fuel-bed temperatures which otherwise may become too high for the successful generation of this gas. Steam, upon meeting an incandescent fuel bed, is decomposed so that the combination of carbon and steam is theoretically broken up into carbon monoxide and hydrogen ($C + H_2O = CO + 2H_2$). The enrichment of the gas from the steam is due to the additional hydrogen. It is also essential that sufficient care be exercised in the use of steam to prevent the chilling of the bed to such a point that the necessary decomposition cannot take place. The amount of steam that can be used to advantage is, therefore, limited. It is possible to maintain combustion with the air which is supplied and at the same time supply such a large amount of steam that its complete decomposition cannot follow owing to lack of temperature in the fuel bed. The result is usually an excess of carbon dioxide and the mechanical mixture of a certain portion of the steam still undecomposed with the gas issuing from the generator.

When coal is used in place of coke there is usually considerable volatile material, especially in bituminous coals, which is distilled from the fresh coal at the top of the producer by the heat in the gas which passes up through the fuel bed. Besides this coal gas there is usually considerable tarry material given off by coal which may or may not be objectionable in the application of the gas according to the method of utilization. If these tarry products or hydrocarbon compounds are allowed to chill they may be very objectionable in certain types of plants owing to their tendency to clog pipe lines, valves, engine governors, etc. It is important, therefore, when this gas is to be used for operating engines that this tar be eliminated from the gas or be itself converted into a gas which may be utilized as a part of the regular output of the plant. The methods of handling these tarry products will be discussed later.

As previously pointed out, carbon monoxide, hydrogen, ethylene and methane are desirable constituents in producer gas. The application of the gas in various industrial uses depends somewhat upon the relative proportions of these different constituents. Gases with a high percentage of hydrogen may be well adapted to certain types of metallurgical application but for power purposes involving the use of the gas in internal-combustion engines it is found necessary to keep the percentage of hydrogen within certain limits. For this reason the methods of operating producer plants for power purposes are often quite different from those applied when the gas is to be used for metallurgical work.

As will be noted in the above analyses, oxygen usually appears in very small percentages. Nitrogen has no special effect but simply acts as a diluent. The third diluent, carbon dioxide, is an undesirable constituent in producer gas and the percentage present should always be the minimum possible with any given grade of coal or method of manipulation. The chief objections to carbon dioxide in the gas are that it indicates the development of more heat than is required in the process of gas making; shows the presence of a larger percentage of nitrogen than would be the case with more perfect operation and also indicates that a certain portion of the carbon monoxide has been burned in the producer or that the thickness of the fuel bed was not sufficient to reduce the carbon dioxide evolved in the incandescent zone to carbon monoxide before leaving the producer. The principle causes of an excess of carbon dioxide are a thin incandescent fuel bed without sufficient depth of carbon to properly decompose the carbon dioxide produced in this incandescent bed and too low temperature in the fuel bed, usually due to an over-supply of steam.

Types of Gas Producers.—Two distinct processes of making producer gas are in use—the up-draft process and the down-draft process. For commercial purposes these processes are applied by different manufacturers in different ways resulting in the following four general types of producers:

(*a*) Up-draft suction producers.

(*b*) Up-draft pressure producers.

(*c*) Down-draft producers.

(*d*) Double-zone producers.

The Up-draft Suction Producer.—As originally manufactured the suction, or reduction of pressure below that of the atmosphere, was produced entirely by the suction stroke of the engine. Today this reduction in pressure is more often produced by the introduction of a mechanical exhaustor, thus simplifying the operation.

The operation of the engine-type suction plant is described below.

As shown in Fig. 229 the essential parts of a suction-producer plant are the gas generator or furnace, the steam generator or boiler, and the gas cleaner or scrubber. A fire is made with shavings, wood, etc., on the grate of the gas generator, the air necessary for combustion being supplied by means of the blower shown at *D*. As soon as the fire is sufficiently kindled the fuel to be used for gas making—charcoal, coke or anthracite—is gradually charged into the producer. The blower for the air supply is driven by hand, or in large plants by electric or other power, until gas of sufficiently good quality to operate the engine is generated. The quality of the gas is roughly ascertained by means of a test-cock at which the gas is lighted. As soon as the test flame shows the right color, which can be readily determined after a little experience, the gas is turned into the engine. The smoke and poor gas developed during the early stages of combustion are discharged into the outside air by means of the purge pipe shown at *E*. As soon as the engine is started, the blowing of

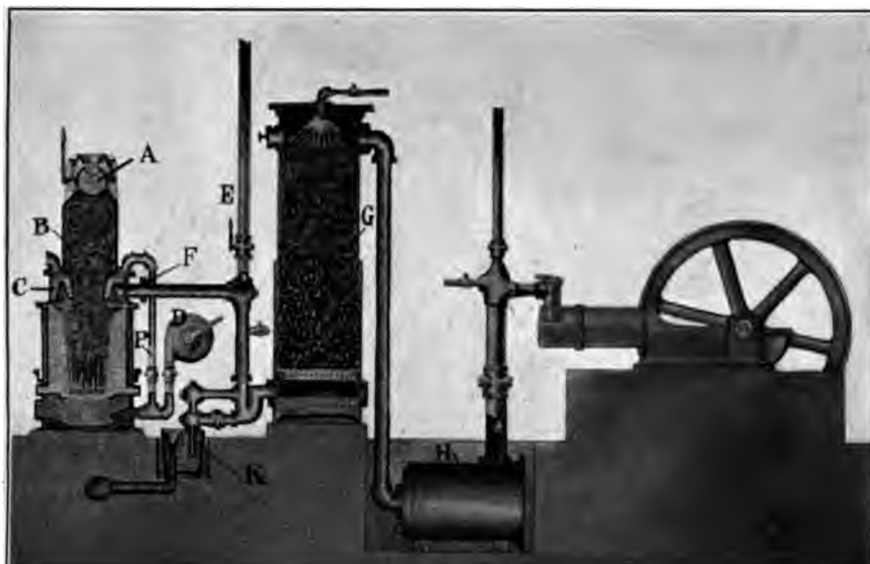


FIG. 229.—Engine-type suction gas producer plant.

the producer is stopped, and the necessary air for maintaining combustion is drawn into the base of the producer by means of the suction produced in the engine cylinder.

If air alone is supplied, even in restricted quantities, the temperature of the fuel bed in the producer rises so high as to hinder the production of satisfactory gas. It is necessary, therefore, as previously stated, to cool the fuel bed by adding steam. The methods of producing this steam vary in detail in plants of different design, but the principles involved are essentially the same.

At *C* is shown the steam generator or boiler for this particular type of producer. Steam at atmospheric pressure is generated by the heated gas, which leaves the producer at the point *F* on its way to the scrubber. The steam thus generated is picked up by the air supply passing into the base of the producer. The mixture of air and steam is then drawn up through the incandescent fuel bed. The oxygen of the air and the oxygen of the steam combined with the highly heated carbon in the lower part of the bed, producing complete combustion and developing carbon dioxide. The gas thus formed passes up through a thick fuel bed above and the carbon dioxide is largely reduced to carbon monoxide. The hydrogen liberated by the decomposition of the steam greatly enriches the product.

Scrubbing the Gas.—The gas after leaving the gas generator at the point *F* passes to the base of the scrubber *G*. The scrubber is usually a

FIG. 230.—Smith gas producer.

cast-iron or sheet-steel tower in which dust, soot, tar and other impurities are removed from the gas. As usually constructed it consists of a simple cylindrical shell filled with coke, over which water is sprayed. The dirty hot gas enters the base of the scrubber and flows upward; it is divided, in passing through the coke, into separate streams which are met by a fine water spray flowing in the opposite direction. The gas and the water are thus brought into intimate contact and the particles of dirt and other foreign matter carried by the gas are largely washed out. The wash water from the scrubber passes into a water seal shown at *K*, from which it overflows into the drain, or, in large installations, into a settling basin or reservoir. If the gas is to be used in an engine it is essential that it be thoroughly freed from gritty material in order to prevent scoring of the engine cylinders. It is equally important that tarry compounds be removed, in order to prevent clogging of the engine valves and governor.

Owing to the fact that the draft of air and steam through the fuel bed is produced by the suction stroke of the engine, it is important that the resistance offered by the fuel bed, scrubber and connections should at all times be a minimum in order that the power from the engine available for commercial use may not be too seriously reduced, by the demands for operating the producer plant itself. With this in view it is essential

Fig. 231.—Smith gas producer plant.

that the grate be kept free from the accumulation of ash and that clinkering in the fuel bed be reduced to a minimum. It should further be borne in mind that owing to this suction action of the engine any tarry products or other foreign matter that may have passed by the scrubber will be drawn directly into the engine valves. Any large amount of tar condensing and cooling in the valves or governor attachments of the engine tends to interfere with its successful operation. For this reason fuels containing large percentages of tar are not available for use in this type of suction producer. This of necessity restricts the fuels in regular use in these installations to charcoal, coke and anthracite coal. Even with certain cokes and anthracite coals there is a slight tar production which has to be properly cared for.

By the introduction of an exhauster of the Root or Connersville type the pressure through the gas-generating system may be maintained below that of the atmosphere without depending upon the suction stroke of the engine. The application of these exhausters is readily seen by reference to Figs. 232, 235, 235A and 237.

The pressure on one side of the exhauster is, of course, negative and on the other positive. This arrangement makes possible the introduction of additional gas-cleaning devices and the possible use of bituminous coal and other tarry fuels.

FIG. 232.—Charging floor Smith producer showing tar extractor.

The largest single unit installed to date is an up-draft suction producer in which the reduction of pressure is maintained by an exhauster as shown by Figs. 231 and 232.

The producer is of simple construction of sheet steel and channels. It is made on the sectional principle and can easily be enlarged by the addition of one or more sections. As operated, the plant shown by the cuts has a fuel-bed area of 210 sq. ft. and burns about 2,750 lb. of Illinois coal per hour.

Owing to the fact that the fuels generally used in the engine-type suction plants are high in price, the installations of this type, although numerous, are of comparatively small power, seldom exceeding 300 hp. per unit and in the majority of cases not exceeding 100.

The Up-draft Pressure Producer.—The pressure producer develops s gas under slight pressure (usually 2 to 8 in. of water). This pressure is ·oduced by means of steam introduced through the blast pipe shown in ıe cut. The air enters the producer through this same pipe, being

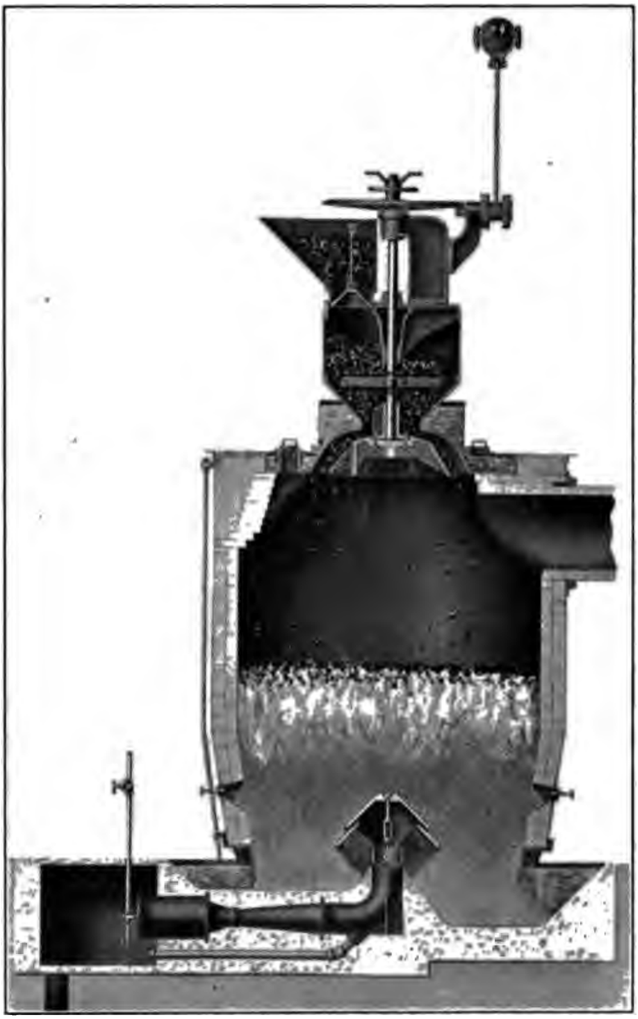


FIG. 233.—Up-draft pressure gas producer.

awn in by means of induced currents produced by the steam. The ɘam is supplied at a pressure of from 40 to 80 lb. by an auxiliary boiler.

In this type of producer it is necessary to carry an ash bed deep enough protect the blast pipe.

On top of this ash bed is the incandescent zone and above this

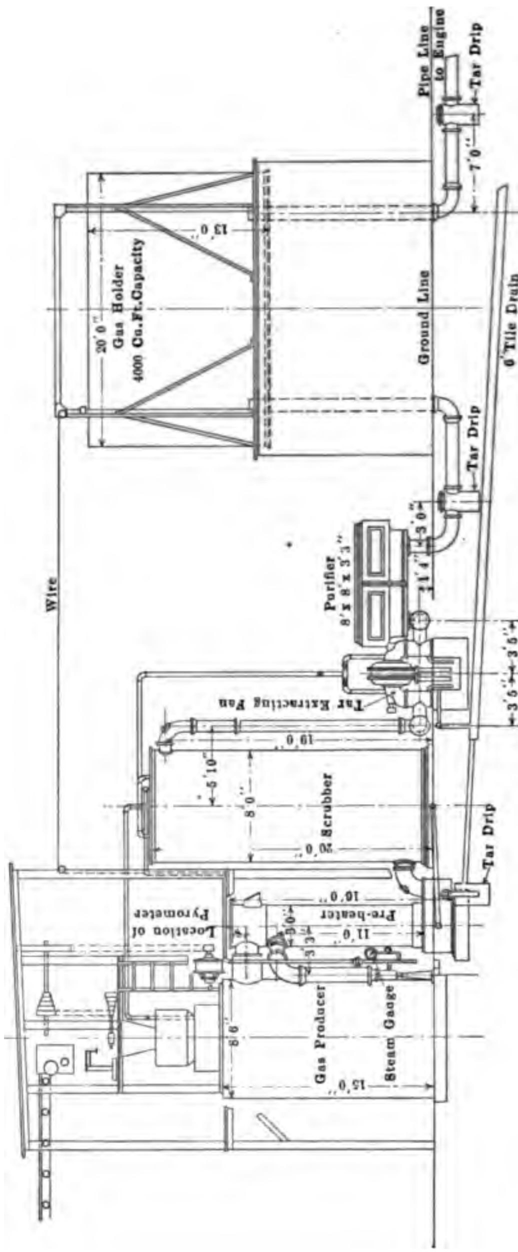


FIG. 234.—Elevation of pressure gas producer.

the deep fuel bed, supplying the carbon for reconverting the CO_2 gas into CO.

The other essential parts of the up-draft pressure producer-gas plant are a preheater for heating the air entering the producer by means of the sensible heat of the hot gas from the producer; a scrubber for cleansing the gas; a tar extractor and tar drips; a pressure regulator; and sometimes a purifier for removal of sulphur from the gas. The elevation of such a plant is shown in Fig. 234.

The Down-draft Producer.—By the mechanical extraction of tar a large part of the heat value of the gas is lost. As already stated, this may not be a serious matter in plants where the sale of the tar for commercial uses brings a good financial return, but in installations where the tar is thrown away the loss is sufficiently serious to warrant the attempt to devise some means of converting this tar into a gas of suitable quality for engine use. Attempts have been made to accomplish this result by manufacturers in this country and abroad, and the success attained has been sufficient to warrant the building of such plants on a commercial basis.

Operation of a Typical Plant. *General Description.*—A typical plant of this character manufactured in the United States is shown in Fig. 235. This plant consists of two gas generators made of steel shells with firebrick linings. As in the types previously mentioned, the essential features of this type of plant are the gas generators or producers, the steam generator or boiler, the gas-cleaning apparatus or scrubbers (both wet and dry), the gas holder or receiver, and the necessary auxiliary piping, etc.

In operating the producer plant the gas generators are charged with coke to a height of 3 or more ft. above the grates, and lumps of coal about 4 in. in diameter are added on top of the coke to the depth about 6 in. A wood fire is then kindled on top of this charge. The valves in the pipes at the base of the producers leading to the boiler being open, the exhauster *a* is started and the entire portion of the plant to its left is thus placed under suction. The air for supporting combustion is drawn in through the charging doors shown above the operating floor at *b*, downward through the fuel bed and pipe line into the base of the boiler, up through the boiler tubes and pipe line to the base of the scrubber, and then up through the scrubber into the exhauster. From the exhauster the gas is under pressure and may be sent through the dry scrubber *c* into the gas holder, or by simply opening the valve may be sent through the purge pipe *d*, into the outside air. When starting the fire, smoke and impure gas are diverted through the purge pipe until such time as the test flame shows the gas is suitable for turning into the holder.

After the coke has become incandescent and the fire in the upper portion of the fuel bed well established, green fuel is added through the

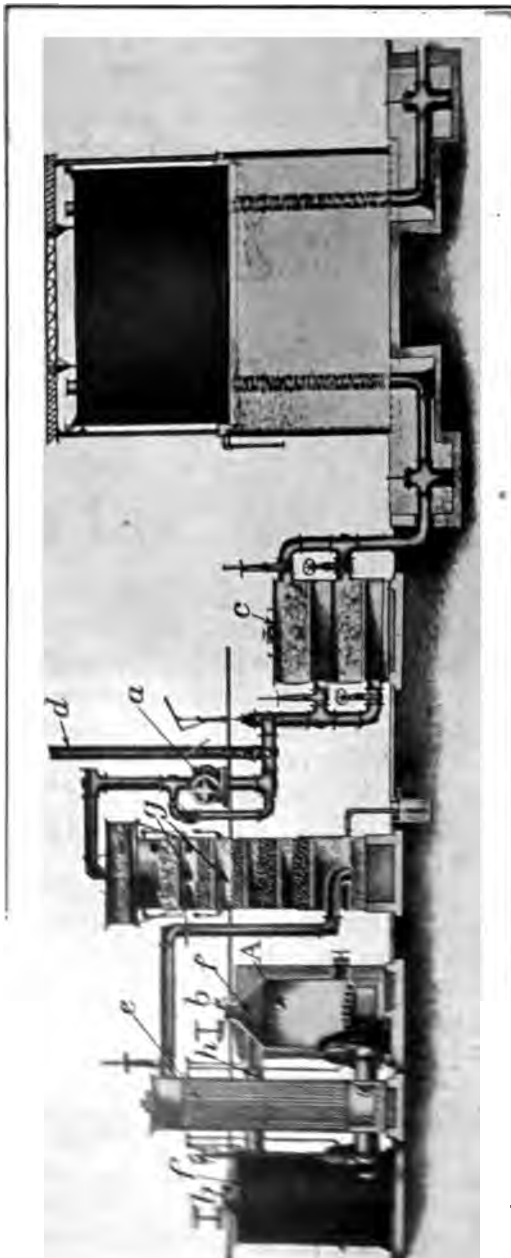


FIG. 235. Down-draft gas producer.

rging doors b as the condition of the fire and the quality of the gas uire.

The steam required by the plant is generated in the tubular boiler e he heat of the gases as they pass from the generator to the scrubber. icient heat is furnished in this manner to supply steam at 60 or 80 ressure in the average plant of this type, although occasionally it is d best to install an auxiliary boiler. The steam which enters the ucer above the fuel bed at the point f mingles with the air entering and the mixture passes down through the fuel bed, as previously ned.

he gas is cleaned by the coke scrubber, although in this particular of plant the scrubber is fitted with partitions or shelves, upon which coke is placed. The water spray which meets the ascending gas ents is shown at g. The upper portion of this scrubber is generally l with excelsior, thus making a dry scrubber which removes consider- moisture from the gas besides impurities. The additional dry scrub- shown at c, is also filled with excelsior, and consists of two chambers, valve connections so arranged that either chamber may be bypassed leaning.

onversion of Tarry Vapors into Fixed Gases.—By this down-draft ess the hydrocarbon compounds and volatile material distilled from green coal in the top of the fuel bed are drawn into the incandescent , where the tarry material is to a certain extent converted into a d gas. The completeness of the conversion is determined by the coal l, the conditions of the fuel bed, and the method of operating. Some- s the process is exceedingly satisfactory and the tarry products are tically all transformed into a good grade of gas; at other times a ion of this material is transformed into gas and a portion is burned he incandescent zone. One of the criticisms of the system is that at e plants much lampblack is produced. Although this may cause no us results so far as the operation of the engine is concerned, yet lamp- k is a disagreeable substance to handle in any quantity and tends to ct in every nook and crevice. If it is not properly removed when s containing a high percentage of sulphur are used there is a possi- y of a weak solution of sulphuric acid forming in certain portions of plant and gradually eating away any iron or steel with which it may e into contact.

'Shooting" the Fuel Bed.—After operating this plant for a few hours he method described, the fuel bed in each of the generators gradually mes clogged with tarry matter, soot, dust, ash, etc., and the suction ired of the exhauster becomes excessive. If the plant is clean this ion amounts to possibly 5 or 6 in. of water, and gradually increases he fuel bed becomes clogged. In the majority of plants it is not deemed wise to allow the suction to exceed about 20 in. of water, although there are plants in daily operation in which 60 in. are carried without difficulty. When this suction becomes excessive, it is necessary to clean the fuel bed. This cleaning, which must be done without stopping the plant, is carried on as follows:

Suppose the fuel bed in the generator marked *A*, Fig. 235, requires cleaning. The charging doors *b*, *b* are closed, then the valve connecting the base of the producer *A* to the boiler is also closed by means of a wheel manipulated from the operating floor, and the steam entering the top of the producer is cut off and steam at full pressure, namely, 60 to 100 lb., is discharged into the base of the producer just below the grate. This high-pressure steam rushing through the fuel bed tends to dispel the tar, soot, dirt, ash, and other foreign material. The process is called "shooting." During the process the current through the fuel bed is, of course, reversed; that is, made to flow upward instead of downward. In the shooting a jet of live steam is turned upon an incandescent fuel bed, and a water gas is formed which is quite different from the gas produced during the down-draft operation of the plant. The heat value of this gas averages about 200 B.t.u. per cubic foot instead of a little over 100 B.t.u., which is the heat value of the gas normally made by the producer. During the process of shaking up and cleaning the fuel bed the water gas is passed up through the fuel bed into the top of the generator *A*; through the bypass pipe behind the boiler, shown at *h*; down through the fuel bed of the generator *B*; and then on through the boiler and scrubber to the exhauster, as in the normal working of the plant. The method of cleaning generator *B* is the same, except, of course, that the operation of the two generators is in reverse order.

Utilization of Water Gas.—The gas produced during the cleaning process, high in hydrogen and of relatively high fuel value, differs so materially from the gas regularly made that many operators of down-draft plants find it inadvisable to mix the two gases, and provide two

DESCRIPTION OF FIG. 235A.

1. Air intake.
2. Producer.
3. Fuel.
4. Waste gas stack.
5. Removable stack.
6. Center charging door.
7. Secondary vaporizer and Water-cooled top.
8. Side charging door.
9. Mixture inlet.
10. Peek hole.
11. Vaporizer.
12. Vaporizer inspection hand hole.
13. Vaporizer water supply.
14. Vaporizer overflow gauge.
15. Top of ash bed.
16. Ash pier concrete.
17. Foundation piers.
18. Waterseal pit.
19. Gas off-take (from prod.).
20. Sprayer column No. 1.
21. Sprayer column No. 2.
22. Water trap.
23. Sprayers.
24. Water supply for trap.
25. Water drain for trap.
26. Up-draft air connection.
27. By-pass connection.
28. Grate for supporting coke in wet scrubber.
29. Wet scrubber.
30. Scrubber sprays.
31. Scrubber gas off-take.
32. Exhauster Del. gas to expansion tank and Engine.
33. Drain.
34. By-pass and up-draft connection.
35. Waste gas stack.
36. Auto. By-pass valve (sending surplus gas back to scrubber).
37. Gas to engine.
38. Combined expan. tank and purifier.
39. Scrubber drain (sealed).
40. Baffle plate.

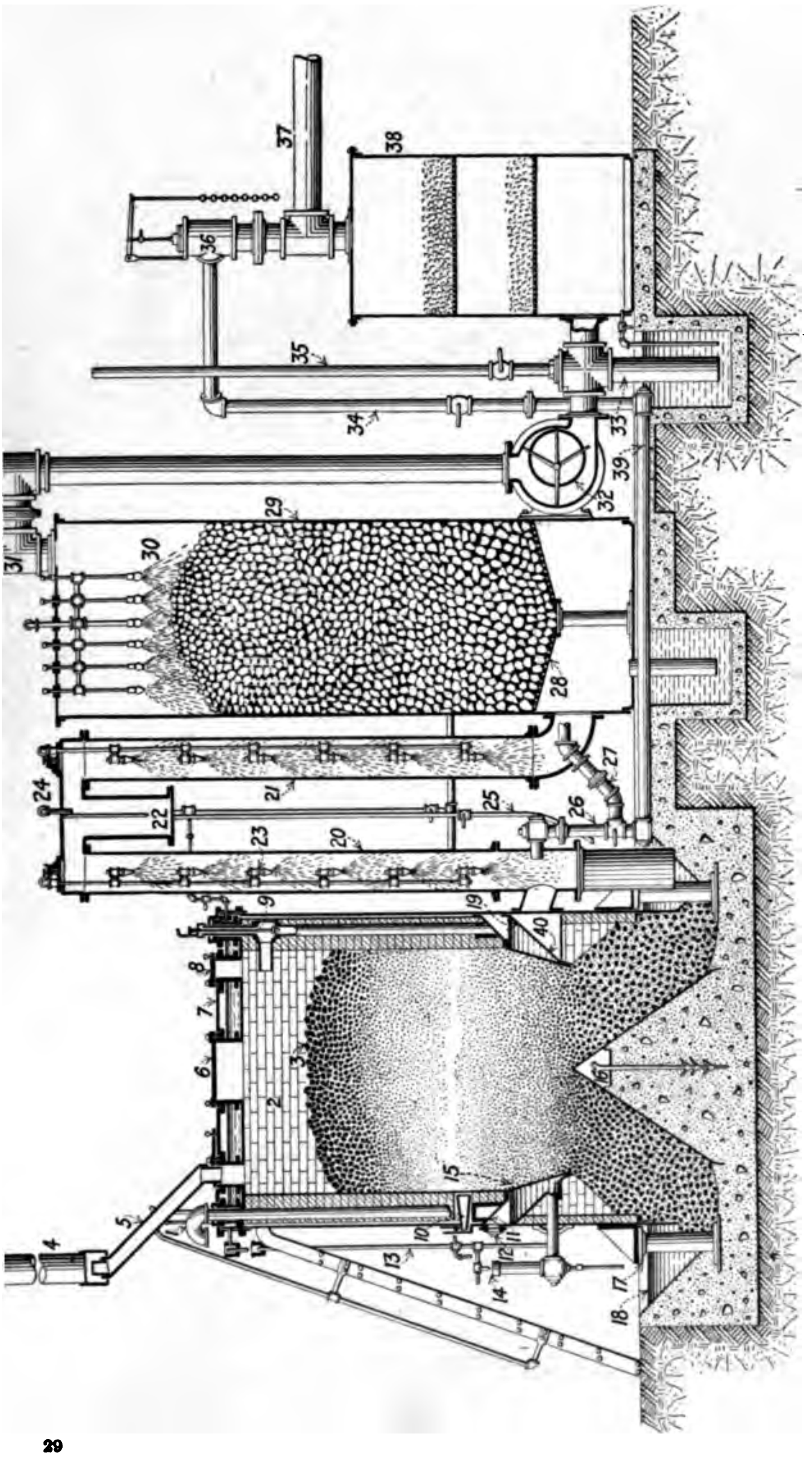


FIG. 235A.—Single generator down-draft gas producer.

gasometers, one for the regular producer gas, or air gas, as it is sometimes called, the other for the gas generated during the shooting process, which is generally termed water gas. The discharge of these two types of gas to their respective gasometers is controlled by the operator on the charg-

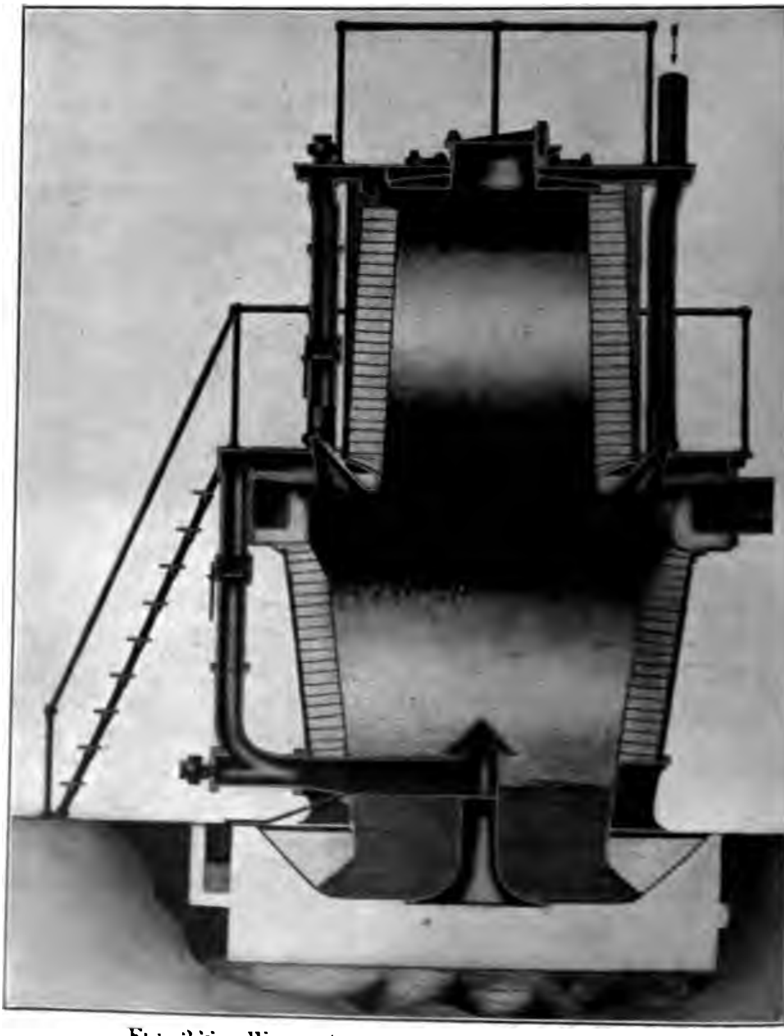


FIG. 236.—Westinghouse double-zone gas producer.

ing floor. The number of times a bed should be shot during the day depends largely upon the character of the fuel used, the demands on the plant, and the methods of manipulation. It is not deemed desirable to utilize the water gas in some plants used only for power purposes, and the gas is discharged into the atmosphere through the purge pipe. In certain plants with high-grade fuels and proper manipulation it is necessary to shoot the bed only two or three times a day. As the period required for the operation is not over a minute or two, the resultant loss from the discharge of gas through the purge pipe amounts to very little. In other plants where producer gas is used for heating, annealing, tempering, forge work, etc., in addition to its use for the development of power, it is advisable to generate as much of the higher heat-value water gas as possible. To this end, the bed is shot as often as may be done without chilling the incandescent zone below an efficient temperature.

Single generator down-draft plants are now in use from which the ash may be withdrawn without shutting down the gas generator thus allowing continuous operation as shown by Fig. 235A.

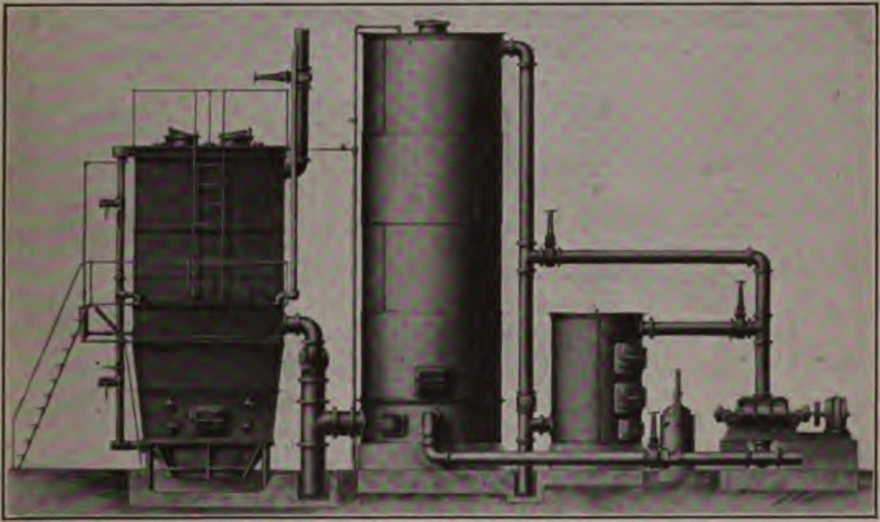


FIG. 237.—Westinghouse double-zone gas-producer plant.

The Double-zone Producer.—The double-zone producer is, as its name implies, a combination of the up-draft and down-draft principles. Two incandescent zones are maintained and the gas is withdrawn at the center or waist-line of the producer. The CO_2 gas formed in both of these zones is thus drawn through the central coke zone where it is reconverted to CO. After the initial fires are started and the plant in operation, the fresh fuel is charged at the top and the volatile matter drawn down through the upper incandescent zone where the hydrocarbons are either burned or converted into a fixed gas, thus destroying the tar. The central coke zone supplies the lower incandescent zone with fuel and in turn is maintained by the coke formed in the upper zone. Care in controlling the distribution of air to the upper and lower zones is required to insure the proper balance for continuous operation.

Vaporizers.—As already pointed out steam is essential in producer-gas making. It is introduced with the air. In many plants it is generated

FIG. 238.—Smith tar extractor. Capacity 250,000 cu. ft. per hour.

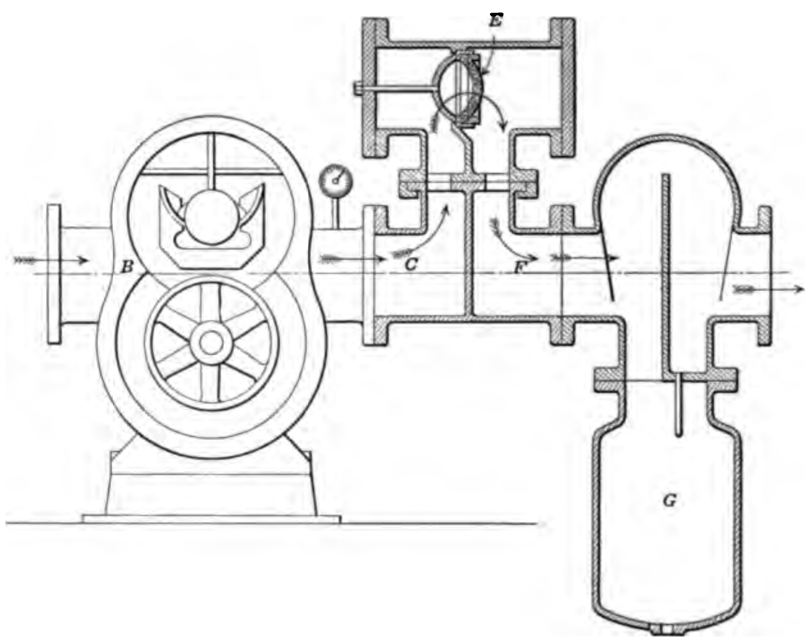


FIG. 239.—Section of Smith tar extractor.

by the sensible heat of the gases as shown in Fig. 229, but for the pressure plants it is usually generated in an auxiliary boiler. For suction plants the steam is at atmospheric pressure or a little below this pressure, while in the down-draft plant of the type shown in Fig. 235, the steam pressure is in the neighborhood of 60 to 80 lb. This is also the usual pressure carried by the auxiliary boilers of the pressure-type installations.

Scrubbers and Tar Extractors.—Ideas regarding the best method of cleaning the gas seem to vary greatly. At one extreme is a scrubber without coke or other solid material, completely filled with finely atomized water or fog, through which the gas passes. At the other extreme is a tall tower-like scrubber with the water pelting down in large drops or globules and supposedly beating the dust and dirt out of the gas. The ordinary practice, however, is to use coke-filled scrubbers and water spray.

In passing through the scrubber the gas is more or less cleansed, depending on the character of the fuel used and the process of gas making. In the down-draft process additional scrubbers filled with excelsior are used for removing moisture and lampblack from the gas. For up-draft

FIG. 240.—Details of a Smith tar extractor.

The above is a photograph of the tar extractor for a 200-hp. producer. The previous illustration shows a section of this apparatus on a plane through the parallel axes of the tar extractor and gas line.

Either of the two extractor heads may be cut in or out of service without affecting the operation of the plant. The heads are ported so that the gas passes through the extractor when its axis is parallel to that of the gas main, and when turned ninety degrees the extractor is out of service. In this illustration the head on the left is out of service and the screens and holder, in which the glass-wool diaphragm is mounted, are dissembled. The coverplate is removed from the head on the right, showing the diaphragm assembled and in place.

plants using tar-producing fuels, some process of tar extraction must be introduced. The most common type of tar extractor removes the tar by centrifugal action, the extractor resembling a centrigufal pump. From the extractor the gas passes through tar drips to water-sealed pits. A liberal supply of water is required for this process, and the speed of rotation of the fan is of vital importance.

A recent form of static tar extractor[1] requiring no water is shown in Figs. 238, 239 and 240.

The descriptive paragraphs are from the manufacturers catalogue.

[1] For a complete description of this tar extractor see *Transactions* A.S.M.E., vol. 35, p. 837.

Special Producer-gas Engine Conditions.—As previously stated, it is necessary in order to make producer gas suitable for use in an engine that it be thoroughly scrubbed and cleaned and that it be sent to the engine at a low temperature. By keeping the temperature low a given volume of gas contains a large number of heat units and, consequently, is capable of developing more power in the engine cylinder than the corresponding volume of the weaker gas. Another advantage in sending the gas cold to the engine is the fact that there is no danger of any condensation of tarry vapors or water vapor after reaching the engine cylinder. This is probably a minor advantage as the engine cylinders are usually kept at a sufficiently high temperature to prevent any possibility of such condensation even if the gas be delivered at a relatively high temperature.

Producers for Metallurgical and Heating Purposes.—Producer gas has for years been extensively used in various types of furnaces in the manufacture of iron and steel. This use has become more and more general during the last few years. In districts where the steel mills have had an abundant supply of natural gas no necessity for a substitute has been felt and no incentive for economizing the fuel supply has existed. The supply of natural gas, however, is by no means unlimited—in some places it has failed altogether—and the time when it will no longer be available in large quantities is near. Large users of this remarkable natural resource have had to recognize these conditions and to hold themselves in readiness to use artificial gas when the supply of natural gas becomes inadequate. The solution of the problem is found in the gas producer, and at the present time there are within the natural-gas regions large installations of gas producers used for the operation of openhearth furnaces.

In the manufacture of producer gas for metallurgical processes the gas goes to the furnace directly from the producer without any cooling or cleaning. It therefore enters the furnace highly heated, carrying with it all volatile hydrocarbons and tarry matter, as gases or vapors, and these add much to the heat value of the gas. The simplicity of the gas-producer equipment required where cleaning of the gas is not essential is shown clearly in Fig. 241. The gas passes directly from the producers to the gas header.

Other Applications as a Fuel.—The abundance of natural gas and the multiplicity of uses to which it has been applied have led to a much greater appreciation of the advantages of gaseous fuel, and have helped to emphasize the value of the gas producer. During the past few years there has been great development in the utilization of producer gas not only for power purposes and in the manufacture of iron and steel, but in other industries as well.

Among the uses to which producer-gas fuel has been put are annealing, japanning, enameling, soldering, brazing, galvanizing, drying, evaporating, tempering, casehardening, type casting, yarn singeing and heating molds, wash kettles, ladles, stoves, bakers' ovens, and cooking. It has

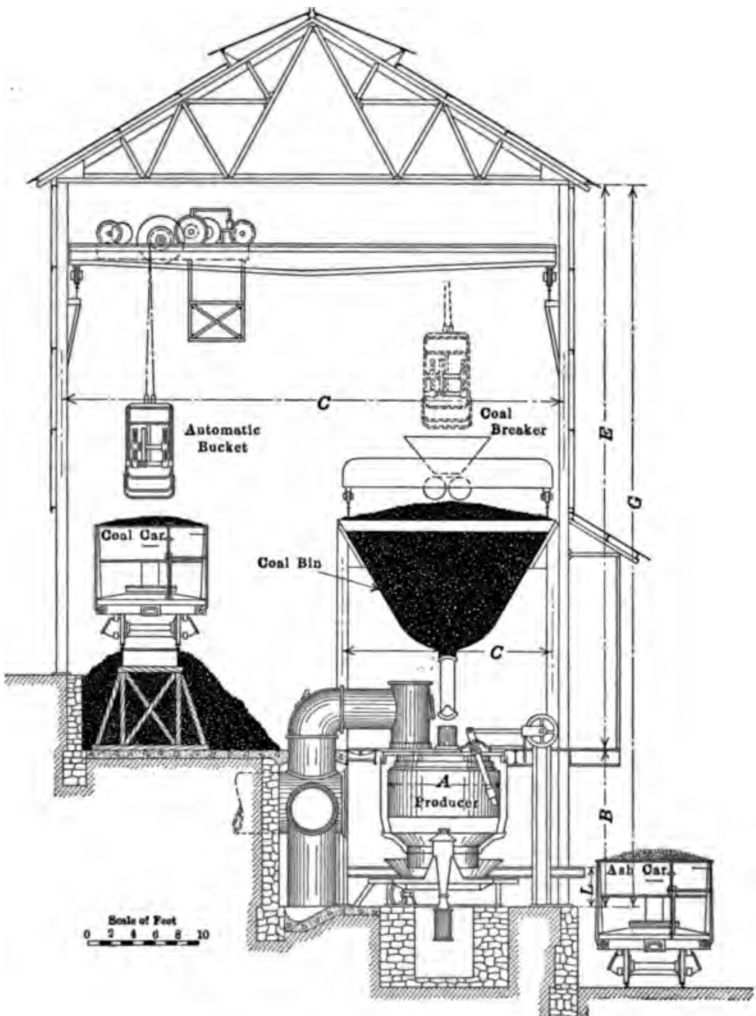


Fig. 241.—Gas-producer plant for metallurgical purposes.

also been used quite extensively in brick, lime and cement kilns, and in various types of ore-roasting furnaces.

Fuels Used.—*Anthracite.*—Little difficulty has been experienced in handling good grades of anthracite coal in gas producers. Occasionally some trouble is experienced due to the character of the ash or to a low ash-fusing temperature. In the main, however, this fuel has been found very satisfactory. For most sections of the country the price of anthracite is relatively too high to warrant its use in plants of large capacity. It is, therefore, largely utilized in plants not exceeding 500 hp. Little has been done in this country with gas producers for the utilization of anthracite screenings or material from the culm pile.

Anthracite coal may be utilized to good advantage in plants of either the up-draft or the down-draft type. Inasmuch as it is comparatively free from tar, anthracite is commonly used in the up-draft producer of the suction type.

A single installation of 4,000 hp. of down-draft producers is using anthracite at $11.30 per ton in preference to bituminous coal for which the plant was designed. Although the company owns bituminous mines, it places a value of $8 per ton upon its books for the bituminous coal. On this basis of $8 per ton for the bituminous coal and $11.30 per ton for the anthracite, a year's operation shows financially in favor of the anthracite. Outside of two or three installations, the individual anthracite plants of this country do not exceed a few hundred horsepower.

Bituminous Coal.—Satisfactory gas producers have been designed for the use of both bituminous coals and lignites of good quality. There is comparatively little difficulty in handling, on a commercial scale, such plants provided the fuel is low in ash, has a fairly high ash-fusing temperature and does not give serious trouble from caking and clinkering. Unfortunately these restrictions are too exacting to fit usual practice in the United States with low-priced fuel. The European situation, where they are able to specify quite definitely the characteristics of the coal, is very different.

The answer to a query as to whether producers have been successfully designed for the use of bituminous coals and lignites, is "Yes" for bituminous coals and lignites of high grades and although it is not "No" for other grades of bituminous coals and lignites, yet it is realized that low-grade fuel, high in ash and prone to clinker troubles, is not regarded in the majority of cases as worth the time and effort required. Bituminous coals and lignites of good grade may be successfully used in the up-draft producer if adequate equipment is installed for scrubbing the gas and removing the tar and in the down-draft producer of the continuous type and in the double-zone producer.

One of the largest single generators in the United States has 210 sq. ft. of fuel bed area burning 2,750 lb. of Illinois bituminous coal per hour. There is no apparent reason why single-shell producers of this type should not be built four times this capacity.

Lignite and Peat.—Both lignite and peat have been successfully used in various types of producer plants. Many commercial installations are operating on the former fuel in this country. Peat has not been commercially developed in the United States, but in Europe it is extensively used as a producer gas fuel.

Amount of Fuel Used by Producer-gas Power Plants in the United States.—An estimate of the horsepower capacity of gas producers in operation for power purposes in the United States in 1915 and the amount of fuel used by these plants is:

	Horsepower
For anthracite coal:	
Plants of more than 500 hp. rating	40,000 hp.
Plants of less than 500 hp. rating	95,000 hp.
For bituminous coal	130,000 hp.
For lignite	15,000 hp.

A corresponding estimate of the annual fuel consumption of these plants is:

Anthracite	240,000 short tons.
Bituminous	400,000 short tons.
Lignite	60,000 short tons.
	700,000 short tons.

Use of Low-grade Fuels.—In the United States the majority of plants are using good-grade fuels, but economic conditions will necessitate before many years use of so-called low-grade material.

Although commercial conditions make reliability of operation and plant capacity imperative, many plants could today utilize to advantage relatively cheap, poor grades of fuel with an assurance of both reliability and capacity and a net financial gain.

The most difficult problem seems to be that of securing thoroughly competent men for the careful supervision of such installations. As indicated below the Bureau of Mines has demonstrated beyond a doubt the possibilities of actually using the following fuels in gas producers.

Fuel from	Variety or size	Per cent. ash	Per cent. moisture	Pounds of fuel, as fired consumed in, producer per b.hp.-hr.
New Mexico	Run-of-mine	19.63	3.62	1.10
Tennessee	Run-of-mine	20.57	3.55	1.45
Iowa	....	20.70	16.69	1.56
Wyoming	....	20.72	9.44	1.70
Wyoming	Run-of-mine	21.73	8.65	1.83
Illinois	Bone	23.12	8.67	2.88
Brazil, S. A.	Run-of-mine	23.44	10.96	2.02
West Virginia	Bone	28.08	2.91	1.26
Pennsylvania	Washery refuse	30.35	2.68	2.34
Pennsylvania	Washery refuse	31.89	2.25	2.76
West Virginia	Bone	43.74	0.47	1.65

Pounds of Fuel per Square Foot of Fuel-bed Area per Hour.—One of the most important commercial items connected with the design and also with the operation of gas producers is the determination of the number of pounds of fuel consumed per square foot of fuel-bed area per hour. This rate of fuel consumption varies radically with different types of plants and with different grades and different types of fuel and has led to much difficulty in designing and in rating producers. Early work in this country followed European practice almost entirely and thereby occasioned a great deal of trouble in properly rating the pioneer plants and brought about the ultimate failure of many of them. Under certain European conditions, in which fuels of a definite grade are specified, high rates of fuel consumption may be obtained. It is not impossible to secure similar rates of consumption under corresponding circumstances in this country, but as selected fuels are seldom obtainable except under test conditions, it has been found that in general in the United States the rate of fuel consumption per square foot of fuel-bed area does not average much over one-half the amount originally guaranteed by early manufacturers. This fact has, of course, led to a decided modification in the design and proportions of many plants.

Returns from the operators of plants throughout the United States indicate the following rates of fuel consumption per square foot of fuel-bed area to be good average commercial practice.

	Anthracite coal		Bituminous coal		Lignite		Peat	Wood
	Avg.	Max.	Avg.	Max.	Avg.	Max.	Avg.	Avg.
Up-draft plants:								
(*a*) Fuel, as fired	10.0	14.0	8.5	14.0	12.0	17.0	15.0	14
(*b*) Fuel, dry	10.0	13.5	8.0	13.0	8.5	12.0	12.0	
Down-draft plants:								
(*a*) Fuel, as fired	....	....	17.5	23.5	26.5	31.5	35.5	
(*b*) Fuel, dry	....	....	16.5	22.0	18.5	22.0	25.5	
Double-zone plants:								
(*a*) Fuel, as fired	....	....	13.5	18.5	21.5	27.0		
(*b*) Fuel, dry	....	....	12.5	17.5	15.0	19.0		

Pounds of Fuel per Horsepower per Hour.—Producer-gas investigations of the United States Geological Survey and the Bureau of Mines conducted with plants not above the average in efficiency showed the following approximate fuel consumption per brake horsepower per hour.

POUNDS PER BRAKE HORSEPOWER PER HOUR

	Bituminous coal			Lignites			Peat[1] avg.
	Avg.	Max.	Min.	Avg.	Max.	Min.	
fired...	1.3	2.0	0.8	2.0	2.8	1.5	2.6
ry......	1.2	1.8	0.8	1.63	2.02	1.35	2.0

;hough these figures were secured during the progress of regular yet the conditions outlined in reports of the Bureau of Mines indilearly that equally good results should be readily secured in the ;e commercial producer-gas plant.

more direct comparison between the results of commercially oper- ılants and those from the Government Station may be had by an tion of the following table.

POUNDS OF FUEL AS FIRED PER BRAKE HORSEPOWER PER HOUR

	Anthracite coal			Bituminous coal			Lignite			Peat[1] avg.	Wood[2] avg.
	Avg.	Max.	Min.	Avg.	Max.	Min.	Avg.	Max.	Min.		
of Mines...	...	...	...	1.3	2.0	0.8	2.0	2.8	1.5	2.6	
rcial plants	1.3	1.5	1.3	1.4	2.4	1.0	2.5	3.0	2.0	...	3.3

relation between the pounds of fuel per brake horsepower-hour calorific value of the fuel may be seen by referring to the following).

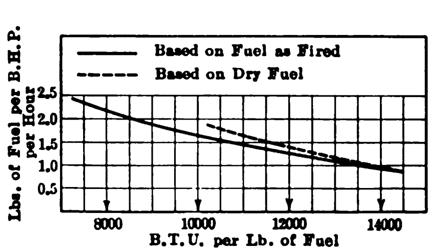


FIG. 242.

tion of Gas.—The composition of producer gas varies with producer, the methods and skill used in operating, the uniregulation of the air and steam supply, the kind and quality the depth of fuel bed, the distribution of the fuel and the size of the fuel.

ɀe of several typical analyses of producer gas from the

[1] ıple of peat only. [2] One sample only.

Bureau of Mines testing plant and the average of the figures presented for plants in commercial operation are as follows:

UP-DRAFT PLANTS

	From anthracite coal		From bituminous coal		From lignite		From peat[1]		From wood[1]	
	Bureau of Mines	Commercial plants	Bureau of Mines	Commercial plants	Bureau of Mines	Commercial plants	Bureau of Mines	Commercial plants	Bureau of Mines	Commercial plants
Carbon monoxide (CO)		22.7	18.28	24.4	21.00	......	21.0	......	......	13.6
Methane (CH_4)		0.0	3.12	3.7	2.30	......	2.2	......	......	8.0
Ethylene (C_2H_4)		0.0	0.18	0.1	0.40	......	0.4	......	......	0.0
Hydrogen (H_2)		15.5	12.90	11.6	18.50	......	18.5	......	......	4.0
Carbon dioxide (CO_2)		5.5	9.84	4.8	12.40	......	12.4	......	......	12.9
Oxygen (O_2)		0.3	0.04	0.6	0.00	......	0.0	......	......	0.6
Nitrogen (N_2)		56.0	55.64	54.8	45.50	......	45.5	......	......	61.7

DOWN-DRAFT PLANTS

	From anthracite coal		From bituminous coal		From lignite		From peat		From wood	
	Bureau of Mines	Commercial plants	Bureau of Mines	Commercial plants	Bureau of Mines	Commercial plants	Bureau of Mines	Commercial plants	Bureau of Mines	Commercial plants
Carbon monoxide (CO)			[illegible]	23.6	15.0	....	14.5			
Methane (CH_4)			[illegible]	[illegible]	1.7	...	1.5			
Ethylene (C_2H_4)			[illegible]	[illegible]	[illegible]	...	0.1			
Hydrogen (H_2)			[illegible]	[illegible]	[illegible]	...	13.3			
Carbon dioxide (CO_2)			[illegible]	[illegible]	[illegible]	..	12.9			
Oxygen (O_2)			[illegible]	[illegible]	[illegible]	.	0.6			
Nitrogen (N_2)			[illegible]	[illegible]	[illegible]		56.5			

OIL-GAS PLANTS[2]

	Fuel, crude oil	
Carbon monoxide (CO)	10.2	7.4
Methane (CH_4)	6.1	12.7
Other hydrocarbons	3.8	2.6
Hydrogen sulphide (H_2S)	0.0	3.1
Hydrogen (H_2)	10.6	
Carbon dioxide (CO_2)	6.1	4.5
Oxygen (O_2)	[illegible]	
Nitrogen (N_2)	[illegible]	69.3

Heat Value[3] of the Gas.—The heat value of the gas from different fuels determined from the average of a large number of tests reported by the Bureau of Mines and also from the figures submitted by the operators of plants in commercial operation are:

[1] One station only.

[2] Records from two commercial plants.

[3] Higher heat values used throughout in these notes.

BRITISH THERMAL UNITS PER CUBIC FOOT OF GAS

	From anthracite coal		From bituminous coal		From lignite		From peat		From wood	
	Avg.	Max.	Avg.	Max.	Avg.	Max.	Avg.	Max.	Avg.	Max.
	Up-draft Plants									
u of Mines.....	...	.....	152	176	158	188	175[a]	....		
nercial plants...	138	.....	151	175	157	185	....	....	133[a]	
	Down-draft Plants									
u of Mines.....	...	.....	110	123	111	127	115[b]	119[b]		
nercial plants...	...	.....	123	130						
	Double-zone Plants									
u of Mines										
nercial plants...	...	.....	(d)	...	118					

Oil-gas Plants

	Gas from crude oil Avg.	Max.
u of Mines		
nercial plants............................	215[c]	230[c]

)ne sample only. [b] Two samples only. [c] Two plants only. (d) Tests indi-
his figure to be approximately 115.

ubic Feet of Gas per Pound of Fuel.—

CUBIC FEET OF STANDARD (60°F. AND 30 IN. HG.) GAS PER POUND OF FUEL
Up-draft Plants

	From bituminous coal				From lignite				From peat			
	As fired		Dry		As fired		Dry		As fired		Dry	
	Avg.	Max.	Avg.	Max.	Avg.	Max.	Avg.	Max.	Avg.	Max.	Avg.	Max.
u of Mines.....	61	101	65	104	36	46	46	53	30[a]	.....	38[a]	
nercial plants...	75	96										
	Down-draft Plants											
u of Mines.....	65	80	68	82	36	44	52	61	29[b]	31[b]	40[b]	44[b]
nercial plants...	79	82										

)ne sample only. [b] Two samples only.

The variations of the gas production with the calorific value of th fuel are shown by the following Bureau of Mines data from an up-dra pressure plant.

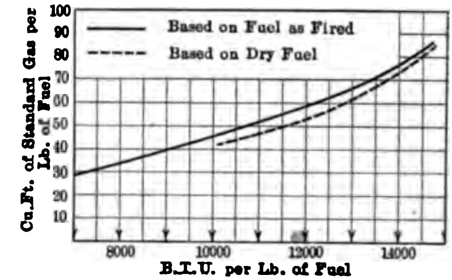


FIG. 243.

On the basis of the average figures from both the Bureau of Mine Testing Station and commercial plants, the following gas yield and hea value may be expected per ton (2,000 lb.) of fuel as fired.

Fuel	Cubic feet of gas per ton of fuel as fired	B.t.u. per cubic foot of gas	B.t.u. in gas per ton of fuel as fired
	Up-draft Plants		
Charcoal	160,000	135	21,600,000
Anthracite	140,000	135	18,900,000
Bituminous coal	135,000	150	20,200,000
Lignite	72,000	155	11,100,000
Peat	60,000	175	10,500,000
	Down-draft Plants		
Bituminous coal	145,000	110	16,000,000
Lignite	72,000	110	7,900,000
Peat	60,000	115	6,900,000

Cleaning the Gas Generator.—Although in some types of produc plants the cleaning out of ashes and clinker is a dirty and tedious job, t majority of plants manufactured today are adapted for continuous a removal so that the labor in operation has been materially lessene Aside from plants of the intermittent character, the labor of cleani seems to depend largely upon the man in charge of the plant.

The variation in the time required for this cleaning, as reported the operators of commercial plants runs from ½ hr. time of one man several hours' time of two or three men. Although some operators ha serious clinker troubles, the majority are now able to avoid this difficult

Time Between Periods of Drawing Fires.—With the exception of intermittent producers in which fires must be drawn every few days—usually from 6 to 15 days—many plants are now practically continuous as shown by the following records of commercial installations.

Hp. of each gas generator	Total hp. of plant	Fuel	Time between periods of drawing fires
250	500	Anthracite	Four years.
300	300	Anthracite	Once a year.
300	300	Anthracite	Indefinite.
400	400	Anthracite	Not drawn.
200	400	Bituminous coal	Once a year.
370	1,100	Bituminous coal	Not drawn except for repairs.
650, 1,000	3,650	Bituminous coal	Once a year.
2,500	2,500	Bituminous coal	Six months to 1 year.
100	100	Lignite	Four months.
200	200	Lignite	Two years.
300	300	Lignite	Once a year.

Power Required by Producer Auxiliaries.—

PERCENTAGE OF TOTAL PLANT POWER

	Installed as auxiliaries			Actually used by auxiliaries		
	Max.	Min.	Avg.	Max.	Min.	Avg.
Up-draft plants.........	15.0	0.7	4.3	6.2	0.7	2.8
Down-draft plants......	10.0	2.0	5.0	10.0	2.0	3.8

Standby Fuel.—Many controversies have arisen regarding the standby losses in producer-gas plants. Two very different values are reported under test and under commercial operating conditions. Several writers on the subject are in the habit of allowing per standby producer hour from 3 to 6 per cent. of the fuel charged in the producer per operating hour.

An attempt was made to secure commercial figures from several plants but the returns are so greatly at variance that no deductions of value can be presented. The figures reported show standby percentages ranging from 3 to 33.

Vaporizer Water Required.—Returns from different operators and tests vary considerably but average figures seem to be:

Fuel	Pounds water per pound fuel fired Up-draft plants	Down-draft plants
Anthracite	0.7–1.0	
Bituminous coal	0.7–1.0	0.23
Lignite	0.0–0.7	0.0
Peat (25 to 30 per cent. moisture)	0.0	0.0

It is possible in some gas power plants to generate the required steam by means of boilers heated by the exhaust from the gas engines. Plants in service are reported as generating from 2 to 3 lb. of steam per horsepower-hour.

Twelve per cent. or more of the power produced in gas engines is available from the waste heat for steam generation.

In large metallurgical operations the amount of steam required in the producer blast averages about 35 or 40 per cent. of the weight of fuel gasified. This is approximately 1 hp. of steam per ton of coal gasified per 24 hr.

Scrubber Water Required.—The scrubber water is rated on the basis of cubic feet per 1,000 cu. ft. of gas scrubbed. In the up-draft plant this includes the water used by the centrifugal tar extractor.

Average figures are:

Bureau of Mines tests, 0.7 cu. ft. for up-draft plants.

Bureau of Mines tests, 10.5 cu. ft. for down-draft plants.

Commercial plants, 14.2 cu. ft. for average for all types of plants.

The high figure reported by commercial plants is undoubtedly due to the fact that in many of these plants water is practically free.

Uses of Tar from Producer-gas Plants.—The indications are that much more general use is made of the tar from producer-gas plants in Europe than in this country.

In England it is used for road making, roofing, briquetting fuel, patent medicines, painting the ends of sawed timber for regulating the drying, etc. In many plants it is converted into pitch.

From 90 to 150 lb. of water-free tar are reported per ton of coal.

The price received varies from $1.25 to $4.25 per ton.

A common method of selling in small plants is 50 cts. per barrel.

One company reports 14.5 lb. of pitch per ton of coal, the tar being converted into pitch.

In the United States a few companies operating producer plants utilize the tar as follows:

(*a*) Fired under boilers in main boiler plant. One plant reports a saving of 5 tons of coal a day, equal to $10.

(*b*) Distilled for oil, creosote and pitch.

(*c*) Run back into gas producers.

(*d*) Mixed with fuel oil for cement burning.

The amount of water-free tar from American fuels seems to run on the average 300 lb. per ton of coal and 100 lb. per ton of lignite.

Mechanically Stirred and Revolving-grate Producers.—Various methods of avoiding difficulties from clinkering and from channels in producer-gas plants have been employed. Among those which tend to do away with hand-poking of the fuel bed are mechanical stirring (see Fig. 244), rotation of the fuel bed in sections and revolving eccentric grates.

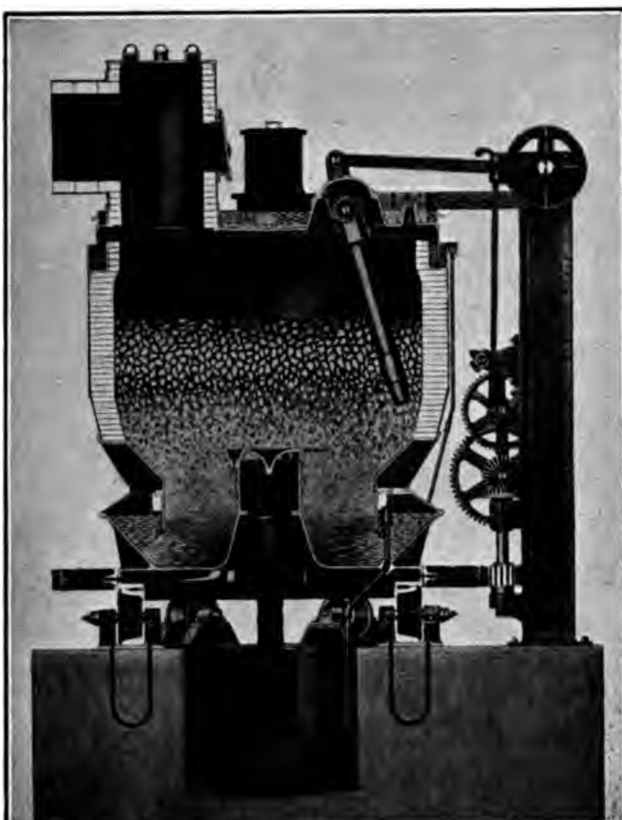


Fig. 244.—Mechanically-stoked producer.

It is claimed by the advocates of these mechanical systems that they of necessity produce more uniform conditions in the fuel bed than can be secured by hand-stoking due to the fact that they are absolutely mechanical and must operate at regular intervals without being subject to the whims or indifferences of the operator. It is also claimed that a much more uniform condition of the fuel bed is secured due to the continual agitation and grinding down of the ash. The uniform fuel-bed conditions claimed naturally tend to produce a more uniform grade of gas and ought to result in a relatively large production of gas per square foot of fuel-bed area.

The opponents of this method of procedure claim that all such mechanical devices are absolutely lacking in judgment and consequently

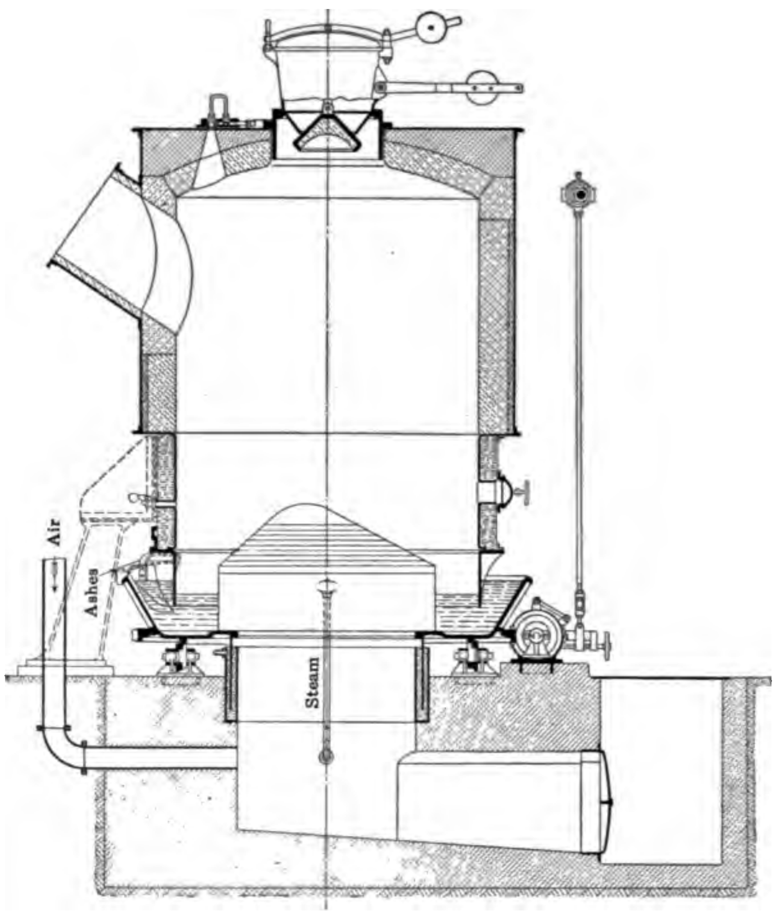


FIG. 245.—Pintsch revolving eccentric-grate gas producer.

do their stoking in a manner which is not adapted to the highest efficiency and most economic conditions required in plants subjected to varying demands. It is also claimed that all such equipment adds materially to the care of such a plant and that this apparatus is forever undergoing repair and failing to respond to the requirements put upon it. It is also claimed that in some of the types of mechanical stokers on the market the waste of fuel by way of the ashes is excessive.

Revolving Eccentric-grate Gas Producer.—The demand for a gas producer to handle all grades of fuel, especially those grades usually sent to the dump, has recently brought to the European market the revolving eccentric-grate producer. This producer appears in several forms, the superiority of each form being firmly established in the minds of its advocates.

The essential features of this producer are shown in Figs. 245 and 246, which show two different types of the eccentric-grate application—the Pintsch and the Rehmann.

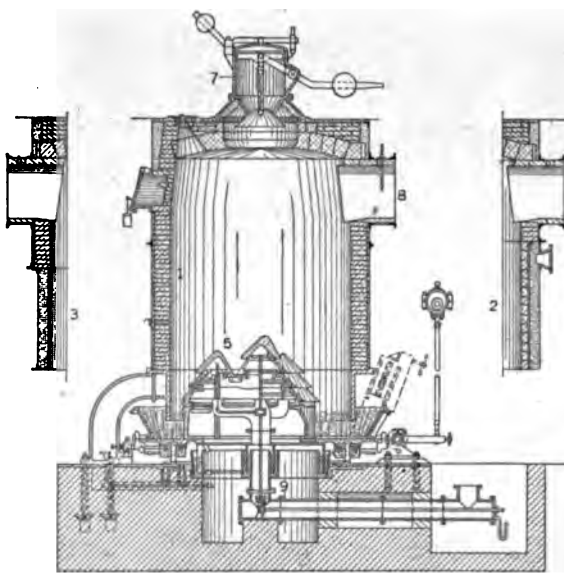


FIG. 246.—Rehman revolving, eccentric-grate gas producer.

Among the most important advantages claimed for revolving-grate producers as compared with the fixed-grate type is constant and automatic ash removal instead of ash removal by hand. Dependent upon this primary advantage rest the following claims for the revolving grate: (1) Low labor cost for handling ashes; (2) more uniform and more complete combustion; (3) operation for months without interruption; (4) ability to handle much more fuel per square foot of fuel-bed area; (5) less space per 1,000 cu. ft. of gas produced or per horsepower of plant; (6) freedom from dust and the usual excessively hot and dirty conditions during removal of ashes; (7) production of a gas of closely uniform quality; (8) reduction in the cost of upkeep.

If, in addition to rotating the grate, the grate be placed slightly off center, a feature is introduced that is probably of far greater value in handling high-ash clinkering fuels than the mere rotation of the grate. The principle of the eccentric grate is clearly shown by Figs. 245 and 246.

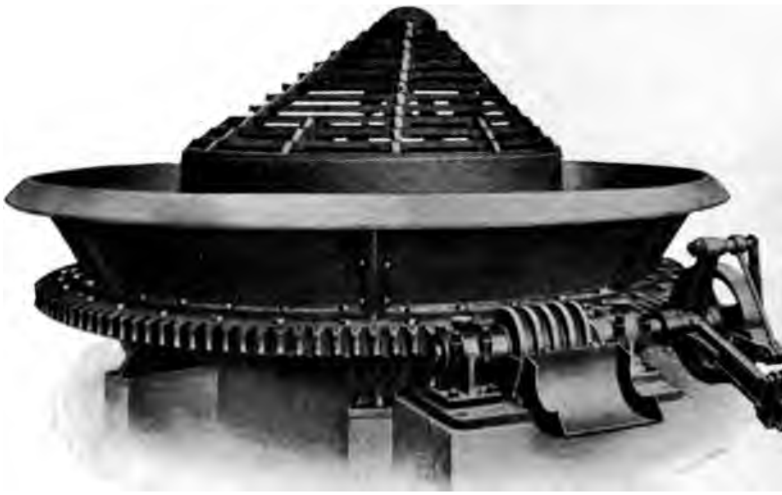


Fig. 247.—Pintsch revolving eccentric grate.

Fig. 248.—Rehmann revolving eccentric grate.

The degree of eccentricity may easily be varied to suit the grade of fuel handled. For fuels that give no trouble from clinkering, or from which the ash is fine, the eccentricity may be reduced to zero, but for uels that give excessive clinkering troubles or from which the ash is coarse the eccentric grate is found of value as it tends to grind the ash in such manner as to prevent the clogging of the system.

The construction of these eccentric grates varies in detail with each patent, as is illustrated by Figs. 247 and 248.

The speed at which the grate revolves is determined by the ash content of the fuel and the demand upon the producer. The usual rate is from ⅛ to 1¼ revolutions per hour. The speed of the grate is so slow that little power is required to drive it. The figure given by the manufacturers is about ¼ hp. for a producer of normal size. The usual practice, however, is to install motors of 1 to 2 hp. for this purpose.

Use of Water Jacket.—Experience with European fuels has shown that even with the eccentric revolving grate and the usual producer-shell construction clinker troubles are not entirely eliminated when a low-grade fuel with low ash-fusing temperature is used. A further important feature—probably the most important single item—for overcoming clinkering and the tendency of the ash to fuse with the producer lining is water jacketing the part of the producer shell surrounding the hot zone. This construction is shown in Figs. 245 and 246.

The extent of this jacketing varies from none for coals that give no trouble from clinker formation or tendency to fuse with the producer lining to a maximum for those fuels that give such clinkering and fusing difficulties.

In certain designs an additional variation is made in the height of the grate to correspond to the clinkering tendency of the fuels used.

Revolving-grate producers are made of either the dry- or wet-bottom type. For extremely fine fuels, such as fine slack and coke breeze, requiring relatively high air pressure for successful gasification, the dry-bottom ashpit is regarded by some manufacturers as being the more desirable on account of the excessive depth of water required by the wet-bottom type.

Advantages of Revolving-grate Producers.—The revolving-grate producers are reported to gasify two to three times as much fuel per square foot of fuel-bed area per hour as can be gasified in corresponding up-draft pressure producers with fixed grates. In the operation of the plants gas leakage is small, as poking of the bed is reduced to a minimum. Work about the producers is thus rendered much more agreeable than is usual with up-draft pressure plants.

Claims of very low percentages of carbon in the ash are also made for this type of producer, the reported record for one installation being per cent. carbon, or 0.47 per cent. of the fuel gasified.

The claims advanced regarding the steam requirements for clinkering coals used in producers with water jackets around the hot zone are to the effect that not more than one-quarter as much steam is required as in the jacketless type with fixed grate. The figures given for comparison are 1 lb. of steam per pound of fuel for the fixed-grate jacketless producer, and 0.25 lb. for the revolving eccentric-grate jacketed producer. Results with United States coals in fixed-grate jacketless producers indicate that 1 lb. of steam per pound of coal is rather high for plants of good size. Seven-tenths of a pound is nearer the figure, although there are undoubtedly many plants, indifferently operated, that are not below the 1-lb. rate.

Efficiency of Gas Producers.—Two efficiencies are usually recognized for producer-gas power plants.

(*a*) Efficiency of conversion and cleaning.

(*b*) Producer-plant efficiency.

By the first of these is meant the ratio of the actual number of heat units in the clean, cold gas delivered to the gas holder or engine to the number of heat units in the dry coal actually charged into the producer for a given time interval. In determining this efficiency no account is taken of the power required to drive any auxiliaries necessary for the manufacture of the gas and no allowance is made for any fuel used in generating the steam required in the blast in plants in which this steam is generated by means of an auxiliary boiler. Under ordinary conditions this is the efficiency upon which the manufacturer bases his guarantee usually expressing it in some such terms as: "We guarantee the gas-generating system to operate commercially when supplied with fuel approved by us and containing not less than .... B.t.u. per pound of dry fuel and when handled according to our instructions to deliver in the form of gas .... per cent. of the heat units contained in the fuel used in the gas generator."

This efficiency, as defined above, relates primarily to power installations in which the gas is used in engines or to other types of plants in which the gas is cooled before being utilized. Under furnace conditions and other applications in which the gas passes directly from the producer to the point of utilization without cleaning or cooling beyond radiation losses, the efficiency of conversion is understood to mean the ratio of the number of heat units in the hot gas direct from the producer to the number of heat units in the dry coal consumed in the producer for any given time interval. This efficiency will usually be considerably higher than the corresponding efficiency for the cool, clean gas as the hot gas usually carries considerable materials in the nature of tarry vapors and gases which add to its heat value but which in cooling are condensed and are carried away in the scrubbing process.

The second efficiency, or that of the gas-producer plant, is the ratio of the B.t.u. in the cold gas delivered to the gas holder or engine to the number of heat units equivalent to the total dry coal consumed in the producer, the total dry coal required by auxiliary boilers for generating steam and any power required for driving auxiliary equipment used in the manufacture of the gas (such as centrifugal tar extractors, cleaners, etc.).

The principal heat losses in gas-producer operation which seriously effect the efficiency are: (*a*) heat in the gases leaving the producer; (*b*) radiation losses; (*c*) heat carried away with the ashes. The first of these is the largest, averaging, according to different investigators, about 10 per cent. of the total heat value of the gas. The temperature of the gas leaving the producer usually runs from 800°F. to 1,200°F.

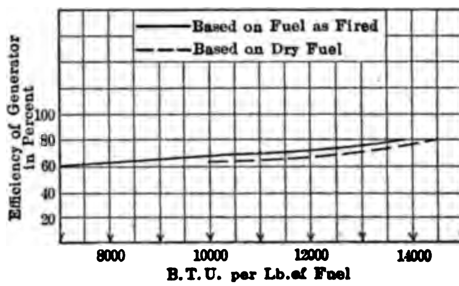


FIG. 249.

Average full-load efficiencies of an up-draft pressure producer operating with different grades of fuel are shown in Fig. 249. In general, the following efficiencies of conversion and cleaning may be used for producer-gas power plants:

Up-draft plants, per cent	70
Down-draft plants, per cent	76
Double-zone plants, per cent	73

For other than normal rating the efficiencies may be taken on the following basis:

Per cent. rated capacity	Per cent. of efficiency at full rating
100	100
75	97
50	92
25	80

Relative Results from Steam and Producer Gas.—In the ordinary manufacturing plant operated by steam power less than 5 per cent. of the total energy in the fuel consumed is available for useful work at the machine. The same figure holds good for average steam locomotive practice. One of the best-designed and most skillfully operated plants in the United States shows only 1,452 B.t.u. delivered in the form of energy to the busbar from each pound of 14,150 B.t.u. coal supplied to the furnace—a thermal efficiency of 10.3 per cent. Probably the maximum efficiency obtained with large unit steam turbines, large unit boilers and exceptional operating conditions is 14 or 15 per cent.—and this but seldom.

On the other hand, results in gas-power practice may equal the following:

	B.t.u.	Per cent.
1. Loss in gas producer and auxiliaries.....	2,500	20.0
2. Loss in cooling water in jackets.........	2,750	22.0
3. Loss in exhaust gases..................	4,250	34.0
4. Loss in engine and generator friction....	315	2.5
Total losses.........................	9,815	78.5
Converted into electric energy..........	2,685	21.5
	12,500	100.0

If advantage be taken of the available heat in the cooling water and in the exhaust gases, a considerably higher efficiency may be realized.

Comparative tests of small plants (250 hp.) of only fair efficiency show the following results for 75 bituminous coals, 6 lignites and 1 peat (Florida):

	Coal	Lignite	Peat
Average ratio, fuel as fired per b.hp.-hr. under boiler to fuel as fired per b.hp.-hr. in producer..................	2.7	2.7	2.3
Maximum ratio for above conditions....................	3.7	2.9	
Minimum ratio for above conditions....................	1.8	2.2	

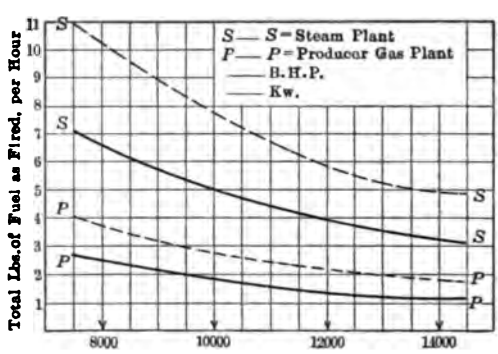


Fig. 250.

The figures for the producer-gas tests include not only the coal consumed in the gas generator but also the coal used in the auxiliary boiler for generating the steam necessary for the pressure blast, *i.e.*, the figures given include the total coal required by the gas-producer plant.

Fig. 250 shows the comparative results in graphical form.

Dimensions of Gas Producers.—An idea of the size of gas producers may be had from the following table, showing approximate relations between horsepower and fuel-bed area for different types of gas generators and fuels.

Type of producer	Fuel used	Horsepower per square foot of fuel-bed area	Horsepower	Approximate inside diameter, inches
Up-draft........	Anthracite and bituminous	8.0 (Up to about 500 hp.)	50	30
			100	48
			200	60
			400	96
			500	108
Up-draft........	Anthracite and bituminous	13.0 (1,000 to 2,500 hp.)	These sizes, as now built, are not circular but made up in sections.	
Up-draft........	Lignite	6.0	50	40
			100	56
			250	86
Down-draft......	Bituminous	13.0 (Up to about 500 hp.) 20 or more for 1,000 hp. and over	100	38
			250	60
			500	84
Double-zone.....	Bituminous	8.5	100	46
			250	74
			500	104

Cost of Gas Producers.—The following table gives the approximate price of suction, pressure, and down-draft producers from 20 hp. to 2,000.

Horse-power	Cost, f.o.b., dollars	Cost of erection, dollars	Foundation, cubic feet	Cost, foundation, dollars	Total cost erected, dollars	Cost per hp., f.o.b., dollars	Total cost per hp., dollars
20	....	....	.....	....	927	.....	46.27
25	....	....	.....	....	1,030	....	42.00
25	650	....	.....	....	.....	26.00	
35	800	....	.....	....	.....	22.80	
50	1,000	....	.....	....	.....	20.00	
60	1,360						
60	1,100	150	50	15	1,265	22.70	
75	1,300	....	.....	....	.....	18.35	21.10
100	1,500	....	.....	....	.....	17.35	
110	1,650	....	.....	....	.....	15.00	
110	....	....	.....	....	1,900	.....	17.30
150	1,850	....	.....	....	.....	12.30	
160	....	....	400	120	3,300	.....	20.60
200	2,450	....	.....	....	.....	12.25	
200	2,500	....	.....	....	.....	12.50	
250	3,000	....	.....	....	.....	12.00	
300	4,300	960	.....	150	5,410	14.35	18.00
500	9,500	....	.....	....	.....	19.00	
1,000	18,000	3,100	.....	....	.....	18.30	
2,000	23,066	3,700	2,140	555	27,321	11.50	13.66

The prices above are from quotations from various manufacturers. It should be remembered that the cost of producer-gas engines is greater per rated power than of engines of the same rating for natural or artificial gas.

Cost of Producer-gas Installations.—Cost per horsepower in dollars.

Horsepower	Gas producer and engine erected, exclusive of foundations	Gas producer and engine erected, including foundations	[1] Complete plant exclusive of buildings	[1] Complete plant including buildings
20	105.00	108.50		
25	62.50	....		
60	69.50	74.50		
75	86.50	....		
80	62.50	....		
110	60.50	68.00		
110	62.00	....		
125	90.00	....	100.00	
250	65.00	68.00	79.00	93.00
500	.....	.....	.....	.....
1,000	55.50	.....	69.50	79.50
2,000	46.00	47.50	56.50	63.50
2,800	.....	.....	.....	76.00
4,000	.....	.....	.....	69.00
4,000	.....	.....	.....	77.50
4,800	.....	.....	.....	72.00
4,800	.....	.....	.....	79.50
5,500	.....	.....	.....	70.00

Relative Cost of Steam and Producer-gas Plants.—A careful study of the relative cost of the two types of installations leads to the conclusion that the complete producer-gas installation for the larger plants—say from 4,000 or 5,000 hp. up—costs about the same as a first-class reciprocating engine steam plant of the same size. A 5,500-kw. installation cost $73 per horsepower for the producer-gas plant. The bid for the corresponding steam plant was reported to be $74 per horsepower.

For plants between 1,000 hp. and 5,000 hp. the gas plant will probably cost from 10 to 15 per cent. more than the corresponding steam plant, but the difference in first cost will be wiped out by the saving in operating expenses within about a year.

For plants below 1,000 hp. the first cost of the producer-gas installation is likely to be from 20 to 30 per cent. more than that of the steam plant. With coal costing $2.75 per ton it will take about 2 years to make up this difference by the saving in operating expenses.

[1] Includes producer, engine, electric generator, and auxiliaries, all erected, with suitable foundations.

In view of the difficulty of determining the exact basis of comparison of the costs of steam and producer-gas plants, the following tables prepared by Mr. Stott of the Interborough Rapid Transit Co., New York City, are presented as indicating the probable relative values of large installations, from the standpoint of maintenance and operation.

A = reciprocating steam engines.
B = steam turbines.
C = reciprocating steam engines and steam turbines combined.
D = producer-gas engine plant.
E = gas engines and steam turbines combined.

OPERATING COSTS OF PRODUCER-GAS POWER PLANT OF THE CHARLOTTE (N. C.) ELECTRIC RAILWAY CO. From *Power*, Apr. 12, 1910

	A	*B*	*C*	*D*	*E*
Maintenance					
1. Engine room, mechanical......	2.57	0.51	1.54	2.57	1.54
2. Boiler or producer room........	4.61	4.30	3.52	1.15	1.95
3. Coal-and ash-handling apparatus	0.58	0.54	0.44	0.29	0.29
4. Electric apparatus.............	1.12	1.12	1.12	1.12	1.12
Operation					
5. Coal- and ash-handling labor...	2.26	2.11	1.74	1.13	1.13
6. Removal of ashes.............	1.06	0.94	0.80	0.53	0.53
7. Dock rental..................	0.74	0.74	0.74	0.74	0.74
8. Boiler room, labor............	7.15	6.68	5.46	1.79	3.03
9. Boiler room, oil, waste.........	0.17	0.17	0.17	0.17	0.17
10. Coal........................	61.30	57.30	46.87	26.31	25.77
11. Water.......................	7.14	0.71	5.46	3.57	2.14
12. Engine room, labor...........	6.71	1.35	4.03	6.71	4.03
13. Lubrication..................	1.77	0.35	1.01	1.77	1.06
14. Waste, etc..................	0.30	0.30	0.30	0.30	0.30
15. Electrical labor..............	2.52	2.52	2.52	2.52	2.52
Relative cost of maintenance and operation..............	100.00	79.64	75.72	50.67	46.32
Relative investment in per cent......................	100.00	82.50	77.00	100.00	91.20

The power house contains two main generating units, each a twin-tandem double-acting Snow engine of 810 b.hp. and a 540-kw. three-phase alternator. The gas producers are the Loomis-Pettibone down-draft type for gasifying bituminous coal. The following operating records and costs were reported Apr. 1, 1910, at the meeting of the American Institute of Electrical Engineers:

OPERATING COSTS OF PRODUCER-GAS POWER PLANT AT WORKS OF A. O. SMITH CO., MILWAUKEE. From *Power*, Sept. 20, 1910

Total engine hours for one year	13,303
Total kilowatt-hours for one year	3,355,907
Total pounds of coal for one year	6,444,281
Pounds coal per kilowatt-hour	1.92
Average load factor	0.45
Coke used in starting producers	260,292 lb.
Coke reclaimed	122,371 lb.
Coke consumed, net	137,921 lb.
Equivalent in coal, as to cost	192,000 lb.
	6,444,281 lb.
Total coal for the year	6,636,281 lb.
Total coal per kilowatt-house, average	1.98 lb.
Cost of coal per kilowatt-hour	0.349 ct.
Cost of power-hour labor per kilowatt-hour	0.170 ct.
Cost of producer labor per kilowatt-hour	0.131 ct.
Oil for power house	0.065 ct.
Oil for producer	0.005 ct.
Waste and sundries, power house	0.012 ct.
Waste and sundries, producer house	0.003 ct.
Repair parts for engines	0.046 ct.
Repair parts for producers	0.007 ct.
Machine-shop work, engines	0.016 ct.
Machine-shop work, producers	0.007 ct.
Excelsior for producers	0.003 ct.
Water, both departments	0.071 ct.
Total cost of power at switchboard per kilowatt-hour	0.885 ct.
Power consumed by auxiliaries:	
Cooling water pump, kilowatts per kilowatt-hour	0.0095
Station lighting, kilowatts per kilowatt-hour	0.0116
Motor-driven exciter, kilowatts per kilowatt-hour	0.0688
Total kilowatts per kilowatt-hour	0.0909

The generating unit of this plant is an Allis-Chalmers tandem double-action gas engine, with 24 by 30-in. cylinders, direct-connected to a 400-kw. D.C. generator running at 150 r.p.m. In the producer room are two 750-hp. Wood producers.

Several kinds of fuel have been tried in this installation, the first being coke breeze at $1.75 per ton, followed by pea size coke at $2.50 per ton. There was found to be a surprisingly large amount of tar in this fuel. The large volume of coke which had to be handled was also an objection and the fuel finally decided upon was pea anthracite, which costs $5.60 per ton delivered. It is stated that 1 ton of the anthracite has been found equal to 3½ tons of the pea coke.

At the time this report was made (September, 1910) the plant had been in operation 10 hr. per day for 5 months, and according to numerous running tests 1 kw.-hr. is being produced from 1 lb. of pea anthracite coal containing 11,500 B.t.u. per pound.

The total cost of the producer and gas-engine installation was $50,000 as against a proposed non-condensing steam installation costing $25,000 for 750 hp., rated capacity of boilers and engines. On this steam plant the builders would not guarantee less than 3 lb. of coal per horsepower-hour running non-condensing and exhausting to the atmosphere. The hot-water heating system, consisting of two locomotive-type boilers and 40,000 sq. ft. of direct coil radiation, was installed at a cost of $19,000, which carries the cost of the entire installation up to $69,000. If the non-condensing steam plant had been installed it was estimated that a vacuum steam-heating system would have cost $30,000, making the total steam installation cost $55,000, or $14,000 less than the present plant investment.

The power plant is now consuming 2,100 lb. of coal for 10 hr. work which, at $5.60 a ton, amounts to approximately $5.90 per day, or a total of less than $1,950 per annum for fuel. Adding to this, the estimated consumption by the heating system, during the coldest months of the year, of 900 tons of steam coal at $3 a ton, makes a total fuel cost for heating and power of $4,650. These figures show a saving of over 60 per cent. per annum in fuel over the estimated performance of a steam plant. Moreover, it was figured that if a condensing plant had been installed, to realize better steam economy, the additional cost of a cooling tower, which would have been necessary because the water is bought from the city, would have materially increased the investment charges for the latter type of plant.

Operating Costs Reported for Other Comercial Plants.

The following division of the items making up the cost of operation of producer-gas plants is taken from two plants in commercial operation:

Plant No. 1	Minimum	Normal	Maximum
Output, kilowatt-hours per day	5,000	8,000	10,000
Fuel, cents per kilowatt-hour	0.25	0.22	0.20
Labor, cents per kilowatt-hour	0.28	0.17	0.14
Supplies and repairs, cents per kilowatt-hour	0.17	0.13	0.11
Operating cost, cents per kilowatt-hour	0.70	0.52	0.45
Fixed charges, cents per kilowatt-hour	0.45	0.28	0.22
Total cost, cents per kilowatt-hour	1.15	0.80	0.67

Plant No. 2	Per cent. total cost	Per cent. operating cost
Fuel	27.4	38.4
Wages	29.5	41.4
Supplies	5.8	8.2
Repairs	8.6	12.0
Operating charges	71.2	100.0
Fixed charges	28.8	
Total cost	100.00 = 0.93 cts. per kilowatt-hour. = $60 per e.hp.-yr.	

Byproduct Producer-gas Plants.—When producer-gas installations are small, not exceeding 3,000 or 4,000 hp., the returns do not warrant any attempt to utilize the constituents contained in the fuel other than the gas itself and possibly the tar. In case, however, the plant is to be of considerable size, say of 4,000 or 5,000 hp., or more, it is possible to not only secure a large supply of producer gas but to separate from the fuel certain byproducts which are commercially of considerable value. The principal installations for byproduct recovery not only produce from bituminous coal large quantities of producer gas but simultaneously with this development yield a liberal quantity of sulphate of ammonia. The figures given for some of the best installations indicate that from each ton of coal there are produced some 80 or 90 lb. of sulphate of ammonia and from 140,000 to 160,000 cu. ft. of gas.

These plants are of necessity somewhat complicated in their operation and are correspondingly high in price so that it is seldom deemed a wise commercial venture to install these byproduct plants when the demand for gas is small.

Anthracite coal does not lend itself profitably to this process owing to its small percentage of nitrogen, and owing to the high initial price of the fuel but the cheaper bituminous coals averaging about 1.3 per cent. nitrogen are especially adapted to the successful operation of such plants provided they are of sufficient capacity to reduce the operating expenses and fixed charges to a reasonable figure per unit of output.

Attention is called to the fact that these byproduct plants are in many cases of large proportions; they are not plants of only 2,000 or 3,000 hp., the majority range from 5,000 to 30,000 hp. One company alone reports the installation of byproduct recovery producer-gas plants using 3,000 tons of fuel a day and aggregating approximately 300,000 hp. The capacity and purpose of a few of the larger installations are as follows:

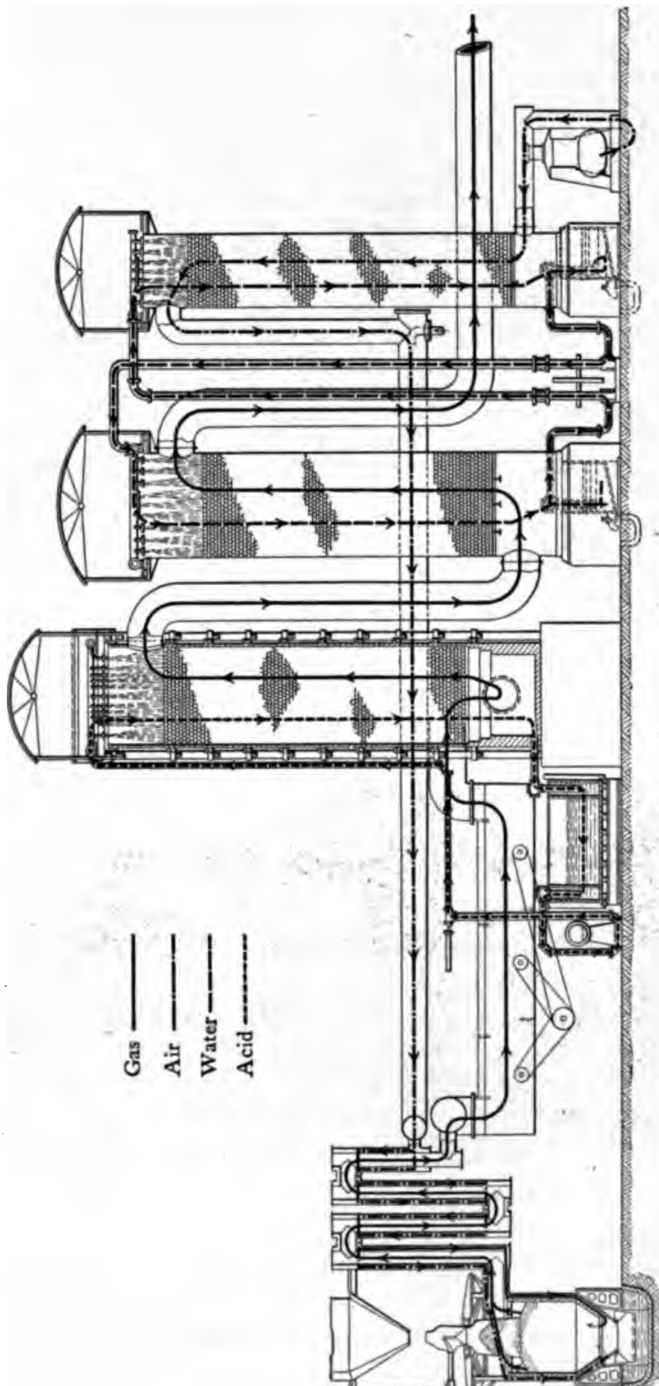


FIG. 251.—Byproduct gas producer plant.

Capacity and Purpose of Large Producer-gas Plants

Installation No.	Fuel capacity per day	Purpose for which plant was installed
	Tons.	
1.......	320	Special plant for the recovery of byproducts from waste fuels. Gas used for firing boilers and for power.
2.......	270	Central distributing station.
3.......	250	Power and chemical purposes, calcining ore, etc.
4.......	150	Special plant for the recovery of byproducts. Gas used for firing colliery boilers.
5.......	135	Power, forge, and plate furnaces, fireclay kilns, etc.
6.......	125	Power and for firing caustic pots.
7.......	120	Evaporating brine.
8.......	120	Power and chemical furnaces.
9.......	100	Firing chrome furnaces.
10.......	100	Chemical furnaces.

Character of Plants.—The majority of these plants are used for power development and gas heating, and the byproducts, such as sulphate of ammonia and tar, are secondary objects in the operation of the plant.

Fig. 252.—Mond Byproduct Gas Plant, at Dudley Port, South Staffordshire, England.

On the other hand, there are several installations in which power is the secondary object, the plant being run primarily for the valuable byproduct, sulphate of ammonia, which brings a commercial return of $50 to $60 a ton.

A few plants are operated for the byproducts alone. In certain districts in which the manufacturing and industrial interests do not offer a market for the gas, the so-called "byproducts" become the main products, and the true byproduct—producer gas—goes to waste. This condition of affairs is peculiarly true in regions in which the local fuel runs high in nitrogen. It is reported that an extensive plant of this character is soon to be erected in Africa.

Peat seems to be peculiarly adapted to the requirements for the production of sulphate of ammonia, and several commercial byproduct plants using this fuel are now in operation in Europe. Among these are two plants in Italy, using 140 tons and 90 tons of peat daily.

The possibilities of a large installation in connection with extensive peat deposits in the United States are now under consideration, considerable preliminary investigation having already been carried forward.

It is reported that many of the peats of this country contain more than 2 per cent. nitrogen and in some cases as high as 3 per cent. As calculated by Davis in his report on "Some Commercial Aspects of Peat as a Source of Chemical Products," he states that the yield of sulphate of ammonia from a short ton of peat containing 2 per cent. nitrogen amounts to 188 lb. and from a peat containing 3 per cent. nitrogen 282 lb. The market price of sulphate of ammonia is approximately $60 per ton which makes the above yields worth $5.65 and $8.45 per ton of theoretically dry peat gasified.

Arthur H. Lymn gives[1] the following operating results and estimates of working costs:

[1] "Gas Producers with Byproduct Recovery," *Journal* of the A.S.M.E., May, 1915.

TABLE I.—ACTUAL OPERATING RESULTS OF POWER GAS PLANT (LYMN SYSTEM) Driving Large Gas Engines and Firing Furnaces

First period of 4 weeks	Total	Average per day of 24 hr., tons	General average
Coal consumption of the gas plant	1,806 tons	64.6	Per kw.-hr. 1.58 lb. (0.72 kg.)
Power produced (kw.-hr.)	1,889,740	.....	Per hr. 2,812 kw.
Yield of sulphate of ammonia	49.11 tons	1.76	Per ton coal 60 lb. (27.1 kg.)
Yield of tar (containing water)	189.7 tons	6.78	Per ton coal 230 lb. (105 kg.)
Average heating value of the gases	155 B.t.u. per cu. ft. (1,380 cal./cu. m.)		
Sulphur contained in the gas (average)	0.63 gram per cu. m.		
Tar contained in the gas (average)	0.04 gram per cu. m.		
The auxiliary machines consumed regularly 71 kw.			
Including 10 per cent. depreciation the gas costs per kw.-hr. work out at 0.069 penny.			
Second period of 4 weeks:			
Coal consumption of the gas plant	1,967 tons	70.2	Per kw.-hr. 1.72 lb. (0.78 kg.)
Power produced (kw.-hr.)	1,899,600	.....	Per hr. 2,830 kw.
Yield of sulphate of ammonia	54.3 tons	1.94	Per ton coal 61 lb. (27.6 kg.)
Yield of tar (containing water)	231.7 tons	8.27	Per ton coal 257 lb. (117 kg.)
Average heating value of the gases	154 B.t.u. per cu. ft.		
Sulphur contained in the gas (average)	0.38 gram per cu. m.		
Tar contained in the gas (average)	0.057 gram per cu. m.		
The auxiliary machines consumed regularly 78 kw.			
Including 10 per cent. depreciation the gas costs per kw.-hr. work out at 0.07 penny.			

NOTE. The nitrogen efficiency during these two periods was 70 per cent. It is frequently 75 per cent.

The working costs of these three plants are based upon the actual results in practice referred to above. It has been assumed that the cost of labor is 50 per cent. and the cost of apparatus is 25 per cent. more in the United States than in England and Germany.

TABLE II.—ESTIMATES OF WORKING COSTS FOR (I) A 2,000-HP. POWER-GAS INSTALLATION; (II) A 4,500-KW. PRODUCER-GAS PLANT, AND (III) A PRODUCER-GAS PLANT FOR CONTINUOUS GASIFICATION OF 500 TONS OF COAL DAILY

Conditions Load conditions of plant	I Power	II Power	III Heating
Hours of full load per annum	4,000	8,500	8,760
Size of plant in b.hp. or kw. or long tons of coal per day	2,000 b.hp. (1,350 kw.)	6,600 b.hp. (4,500 kw.)	500 tons
Cost of coal in dollars per short ton	2	1	2
Heating value of coal in B.t.u. per lb	12,600	12,600	12,600
Nitrogen content of coal in per cent	1.3	1.3	1.3
Cost of sulphuric acid (140° Twaddell) in dollars per short ton	9	9	9
Value of sulphate of ammonia in dollars per short ton	55	55	55
Value of tar in dollars per short ton	5	5	5
Heat consumption of gas engines in B.t.u. per kw.-hr	14,900	14,300	
Cost of plant			
Producer power gas and ammonia recovery plant (Lymn system) in dollars	40,600	126,500	605,000
Buildings and foundations for same	4,400	12,000	55,000
Complete gas engine installation consisting of gas engines, dynamos, all auxiliary machines, exhaust boilers, overhead crane, etc., in dollars	88,000	335,500 (spare set of 2,250 kw.)	
Buildings and foundations in dollars	13,000	48,000	
Total cost of installation	138,000	522,000	660,000
Working data			
Amount of kw.-hr. per annum	5,400,000	38,250,000	
Tons of coal used (including standby losses) per annum	4,830	29,840	204,400
Tons of sulphate of ammonia recovered per annum	206	1,346	9,210
Tons of tar recovered per annum	230	1,500	10,500
Tons of sulphuric acid consumed per annum	190	1,280	8,800
Rate of amortization on machines and plant in per cent. per annum	12	12	12
Rate of amortization on buildings and foundations in per cent. per annum	6	6	6

TABLE II.—*Continued*

Conditions Load conditions of plant	I Power	II Power	III Heating
Annual working costs in dollars of producer gas and ammonia recovery plant (Lymn system).			
Cost of coal	9,660	29,840	408,800
Labor	5,600	16,630	49,500
Repairs and maintenance	1,230	3,780	18,000
Oil, waste, lighting, etc	680	2,990	15,330
Sulphuric acid	1,710	11,520	79,200
Depreciation and interest	5,132	15,900	75,900
Total debit	24,012	80,560	646,730
Credit by sulphate of ammonia	11,330	74,030	506,550
Credit by tar	1,150	7,500	52,500
Total credit	12,480	81,530	559,050
Total annual cost of gas	11,012	970 Profit	87,680
Cost of gas in cents per 1,000 cu. ft. (heating value 150 B.t.u. per cu. ft. m.)	2.10	0.03	0.32
Annual working costs of gas-engine plant. (Based upon first-class German gas-engine practice)	Dollars per annum	Dollars per annum	
Cost: Cost of gas as above	11,012	970 Profit	
Repairs	1,250	5,170	
Oil, waste, water	840	4,420	
Labor at American rates	3,590	10,370	
Depreciation and interest	10,380	3,180	
Total costs	27,072	62,170	
Total cost of power in cents per kw.-hr	0.50	0.16	
Total cost of power in dollars per kw.-yr	....	13.80	
Total cost of power in dollars per hp.-yr	....	10.30	

Slagging Gas Producers.[1]—Many attempts have been made to develop a gas producer along blast-furnace lines in which the ash should be fused and drawn away as a molten mass.

Little of commercial value has been accomplished along this line in the United States, but several such plants are operating with more or less success in Europe. Such a plant in Deutsch-Luxemburg inspected by

[1] For detailed descriptions, see Bureau of Mines *Technical Paper* No. 20.

one of the authors in 1914 consisted of four slagging producers approximately 10 ft. in internal diameter and gasifying 60 tons of fuel each per 24-hr. day. The following details are of interest:

Charge.—Thirty kilos of blast-furnace slag to 1 cu. meter of coke breeze.

Blast.—Preheated. Preheated blast has proved much more efficient than cold blast, so additional preheaters are being installed.

Slag.—The slag from the producers is used for brickmaking.

Iron.—Four hundred kilos of iron are recovered as a daily byproduct from 60 tons of fuel.

Byproduct Coke-oven Gas Plants.—Another gas that has been used more or less extensively and successfully abroad and in one or two in-

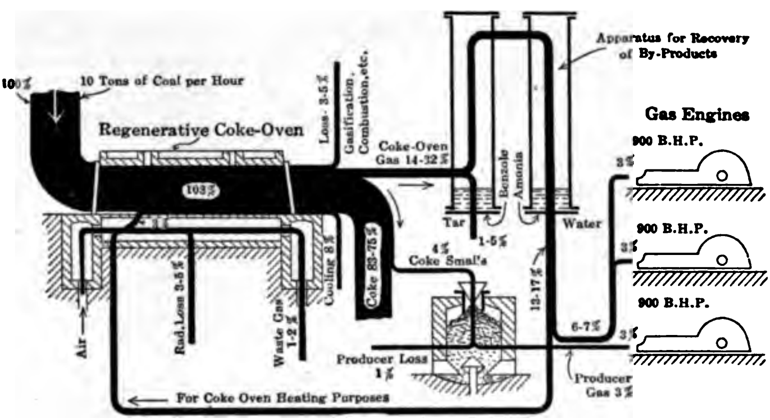


FIG. 253.—Diagram showing utilization of coke-oven waste gases.

stallations in this country is the gas from the byproduct coke-oven plant so that the byproduct coke oven in a sense may be regarded as a gas producer. This method of gas generation has attracted considerable attention of late and in the minds of some is to become so important that the erection of large byproduct coke-oven plants will go on extensively. It is claimed that by this process sufficient coke can be secured for practically all domestic requirements and that coke thus sold will pay the operating cost of the installation and that the gas may be regarded as an extra or byproduct. Others claim that the gas is becoming so important that it will pay to erect such plants for the value of the gas and that the coke will be the extra or byproduct, the gas paying for the cost of operating the entire installation. In other words, the opinions seem to be that either will pay the operating expense of the plant and the other is the profit making byproduct.

Gas Distribution.—One of the special advantages of gas, both as fuel and for power generation in a gas engine, is the ease with which it may be piped to different portions of the plant. If the gas is properly cleaned and freed from tar it ought to cause no trouble in the distributing lines. For such local distribution the pressure can be very low, usually about 5 lb. per square inch for long-distance transmission, thus making the upkeep of the pipe system small and leaks can easily be cared for. There is nothing to condense in the pipe lines so no care is required in the way of insulation.

The situation is totally different in the case of steam distribution. Here the pressures are high (100 to 150 lb. per square inch). The repair bill is often excessive and leaks are a constant source of trouble. Insulation is required and adds to the cost and the methods of pipe laying are necessarily expensive. Serious losses result from condensation in case the steam lines are long and improperly insulated. The pressure in the lines must always be high as the steam must enter the engine cylinder at or a little above the initial pressure required in the cylinder. For a gas engine the only pressure required in the line is enough to insure the delivery of the gas, and in many cases the gas is drawn into the engine by the suction stroke and is at a pressure below that of the atmosphere.

Troubles from condensation and leakage and the cost of installation usually force the erecting of a large steam engine close to the boilers in order to limit these difficulties. With the ease of gas distribution indicated the gas engines may readily be located at various portions of the works.

The distribution of producer gas over extended areas is destined to become a considerable enterprise.

The Blast Furnace as a Gas Producer.—One of the most common types of gas producers is the blast furnace. It is, of course, not built primarily for gas production and the gas is a byproduct of the iron manufacture. Its composition is approximately:

	Per cent.
Carbon monoxide	23
Carbon dioxide	12
Hydrogen	2
Methane	2
Water vapor	3
Nitrogen	58
	100

Calorific value, [illegible] to [illegible] B.t.u. per cubic foot.

As delivered by the furnace the gas contains from 3 to 10 grains of dust per cubic foot of the gas. For engine use the dust content must not exceed [illegible] grain per cubic foot.

Blast furnace gas is usually cleaned in three stages:

1. Dry cleaning to 1.5 to 2 grains per cubic foot.
2. Static washing to about 0.15 grain per cubic foot.
3. Mechanical washing (usually by Theisen washers) to 0.015 or less grains per cubic foot.

The Theisen washer according to Sampson[1] requires about 3 per cent. of the output of the blast-furnace gas output and 16 to 18 gal. of water per 1,000 cu. ft. of gas cleaned. The static scrubbers require 75 to 80 gal. per 1,000 cu. ft. of gas cleaned, making the total 90 to 100 gal.

It is claimed that new processes of gas cleaning require only 20 gal. of water per 1,000 cu. ft. of gas.

Mr. H. J. Freyn presents the following calculations which show the amount of gas available for power generation in blast-furnace plant

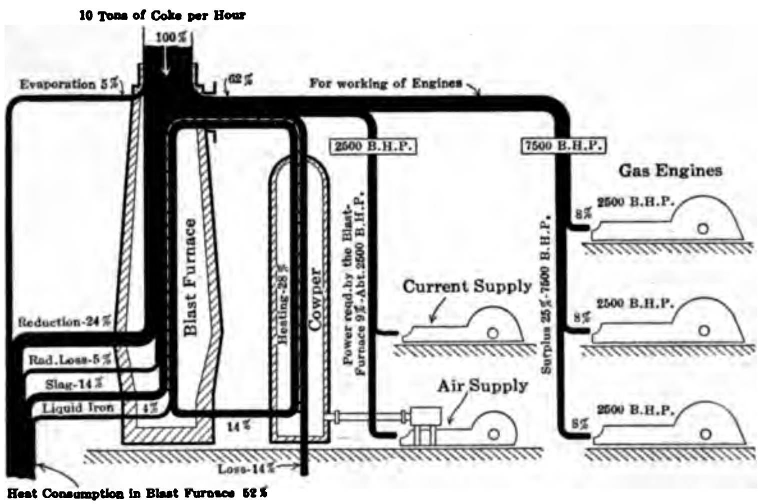


FIG. 254.—Diagram showing utilization of blast-furnace waste gases.

practice. Mr. Freyn's figures are based on a plant of eight furnaces of about 450 tons capacity each, producing about 3,600 tons of pig iron per 24 hr. The amount of blast-furnace gas generated in this plant will be 22,500,000 cu. ft. per hour, since it is generally agreed that 150,000 *cu. ft. of gas are liberated per ton of pig iron produced per* 24 *hr.* This gas will have an average heat *value of* 95 *B.t.u. per cubic foot.* Of this total quantity, *about* 40 *per cent. or* 9,000,000 *cu. ft. is used for heating the blast in the stoves or lost by leakage.*

For gas-blowing engines, about 2,600 brake horsepower per furnace are required, which consume, at the rate of 12,000 B.t.u. per brake

[1] "Practical Operation of Gas Engines Using Blast-furnace Gas as Fuel," C. S. SAMPSON, *Transactions* A.S.M.E., vol. 35, p. 151.

horsepower-hour, about 330,000 cu. ft. of gas per hour, or for eight blast furnaces 2,640,000 cu. ft. An additional quantity of 460,000 cu. ft. per hour for eight blast furnaces will be necessary to produce in gas-electric engines the required power to operate the furnace auxiliaries, such as air compressors for mud guns, skip hoists, ore-handling machinery, transfer cars, bells, lighting, and so on.

The total quantity of blast-furnace gas which has to be deducted amounts thus to 12,000,000 cu. ft. per hour. In other words, there will remain for use, outside of blast-furnace operation, 10,400,000 cu. ft. per hour. This quantity of gas represents at a heat value of 95 B.t.u. per cubic foot the total amount of heat of 1,000,000,000 B.t.u. per hour.

To make use of this available quantity of heat for power generation, gas-electric engines or steam turbo-generators can be installed. In the former case, the available quantity of heat will produce at the rate of 16,200 B.t.u. per kilowatt-hour at the switchboard, corresponding to an average thermal efficiency of 21 per cent., a total of about 60,000 kw. (90,000 b.hp.).

If this available quantity of heat is converted into power through gas-fired boilers and steam turbo-generators, the maximum capacity of the power plant would be about 30,000 kw. (45,000 b.hp.) if a heat consumption of 32,500 B.t.u. per kilowatt-hour, or a thermal efficiency of 10.5 per cent. of the steam plant is assumed.

Cost of Blast-furnace Gas Electric Power Plants.—The cost of these installations apparently runs from about $80 to $105 per kilowatt of maximum continuous rating.

The reported cost of five such installations is:

Power plant No.	1		2		3		4		5	
No. of units	17		2		4		4		5	
Cap. kw., max. con. rating	40,000		4,500		9,000		9,000		11,400	
Cap. b.hp., max. con. rating	56,400		6,400		12,800		12,800		16,300	
Cost of installation per kw., max. con. rating		%		%		%		%		%
(a) Buildings	$9.87	11.3	$75.50	81.8	$10.17	10.6	$10.90	10.8	$10.52	10.3
(b) Eng. equipment	71.78	82.0			72.75	74.6	77.78	76.4	80.32	77.3
(c) Gas cleaning plant	5.85	6.7	16.80	18.2	14.40	14.8	13.00	12.8	12.76	12.4
Grand total, power, plant, complete, per kw	$87.50	100.0	$92.30	100.0	$97.32	100.0	101.68	100.0	103.60	100.0

One large installation of this character is reported to have a cost a little more than $75 per kilowatt.

Cost of Blast-furnace Gas-electric Power.—The cost of producing electric power by means of blast-furnace gas engines in one of the large steel plants is reported to be as follows:

COST OF PRODUCING ELECTRIC POWER. ALL COST FIGURES IN CENTS PER KILOWATT-HOUR

	1910		1911		1912	
Capacity in kw	40,000		40,000		50,000	
Kw.-hr. produced	116,535,000		157,742,510		286,575,000	
Use factor, per cent	33.3		45.0		64.5	
Cost of installation.						
Per kw	$88.00		$88.00		$88.00	
Labor	0.0678		0.0421		0.0302	
Repairs and maintenance	0.0366		0.0305		0.0273	
Lubricants	0.0116		0.0100		0.0085	
Water	0.0074		0.0057		0.0036	
Miscellaneous	0.0064		0.0153		0.0128	
		%		%		%
Total net op. exps	0.1298	17	0.1036	17	0.0824	17.5
Value of gas	......	.....	0.1508		0.1464	
Cost of purification	......		0.0219		0.0144	
Total cost of purified gas	0.1951	25	0.1727	28	0.1608	34.0
Operating cost without fixed charges	0.3249	42	0.2763	45	0.2432	51.5
Fixed charges at 15 per cent	0.4520	58	0.3360	55	0.2310	48.5
Grand total at switchboard	0.7769	100	0.6123	100	0.4742	100.0

PROBLEMS

79. With Pennsylvania bituminous coal, what would be the internal diameter of a single-generator up-draft producer to supply gas for an engine of 500 hp.?

80. (*a*) With a high-grade West Virginia coal, what would be the internal diameter of a single-generator up-draft producer to supply gas for an engine developing 500 hp. when running at normal full load?

(*b*) If the engine were developing only 250 hp., how many pounds of fuel would be burned per square foot of fuel-bed area per hour?

81. A lignite mine is estimated to contain 1,000,000 tons of lignite as mined. How many gas-producer installations of 2,500 hp. each can this mine supply on the basis of a 12-hr. day and full load for 308 days per year for a period of 50 years? What would be the fuel-bed area of:

(*a*) Up-draft producers.
(*b*) Down-draft producers.
(*c*) Double-zone producers.

82. (*a*) Given a 300-hp. producer-gas engine running at 80 per cent. of rated full load; a 100-hp. engine running at 50 per cent. of rated load; an 80-hp. engine running at 60 per cent. rated load. Determine the diameter of one up-draft pressure producer to supply gas for the plant based on average figures allowing for a demand on the producer of 30 per cent. above that indicated by the above loading. Coal, bituminous 12,800 B.t.u. as fired.

(b) If a two-generator down-draft unit be installed instead of the up-draft unit, what would be the internal diameter of each generator?

(c) What would be the main fuel-bed area for a double-zone producer for the same purpose.

83. Determine the amount of water required by the entire plant, engines and producers, of problem 82.

84. Some of the features of a contract for an up-draft pressure gas-producer installation for developing full load with a 750-b.hp. engine are: with bituminous coal of 13,500 B.t.u. per pound, not exceeding 10 per cent. ash, the engine will not require above 60,000 cu. ft. of gas per hour; the engine is to run on the basis of 11,000 B.t.u. per brake horsepower per hour at full load; the internal diameter of the producer is to be 11 ft.

(a) Can the coal indicated be used in the type of producer specified? If so, what becomes of the tar? If not, why not?

(b) Is the fuel-bed area large enough?

(c) Is the guarantee for number of cubic feet of gas a safe one?

(d) If a two-generator down-draft unit be installed instead of the up-draft unit, what would be the internal diameter of each generator?

(e) If a steam plant is put in instead of the proposed gas plant, how much will the cost of coal for the steam plant exceed that for the gas plant per month of 25 days, 10 hr. per day neglecting standby losses, if the coal costs $2 per ton (2,000 lb.).

85. An up-draft pressure producer designed to furnish gas from a 13,500 B.t.u. bituminous coal to a 350-hp. engine was recently installed. The price of coal has increased and the purchaser is considering the use of lignite or peat in place of coal.

What engine capacity could be maintained with these fuels?

86. What would be the probable total difference in cost of bituminous coal at $2.75 per short ton for developing 350 b.hp. for an 11-hr. day, 308 days per year: (including standby losses for 365 days—308 working days and 57 Sundays and holidays), in a pressure gas-producer plant and a steam plant consisting of water-tube boilers and a compound-condensing, high-speed engine using coal averaging 13,500 B.t.u. as fired?

CHAPTER XXIII

COMPARATIVE EFFICIENCIES AND OPERATING COSTS FOR DIFFERENT TYPES OF INSTALLATIONS

Thermal Efficiencies of Different Types of Engines.—

1. Ericcson Hot Air Engine.
2. Direct-acting Steam Pump.
3. Simple Automatic, Non-condensing.
4. Simple Corliss, Non-condensing.
5. Compound Automatic, Non-condensing.
6. Simple Automatic, Condensing.
7. Simple Corliss, Condensing.
8. Compound Automatic, Condensing.
9. Steam Turbine, pres. and vac. high.
10. Compound Corliss, Condensing.
11. Compound Pumping Engine, Condensing.
12. Compound, Moderate Superheat.
13. Compound, High Superheat.
14. Triple Expansion Pumping Engine.
15. Steam Turbine, pres., vac. and superheat high.
16. Quadruple Expansion Pumping Engine.
17. Gas Engine.

The graphic chart above shows the relative thermal efficiencies of the different engines mentioned as reported by good authorities. The thermal efficiencies are based on the indicated horsepower and are the maximum officially recorded.

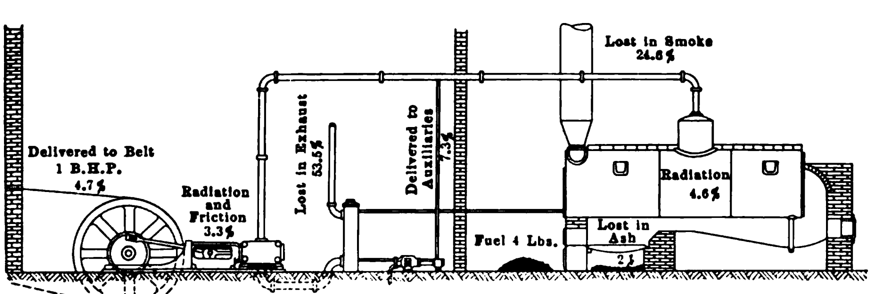


Fig. 255.—Heat distribution—steam plant.

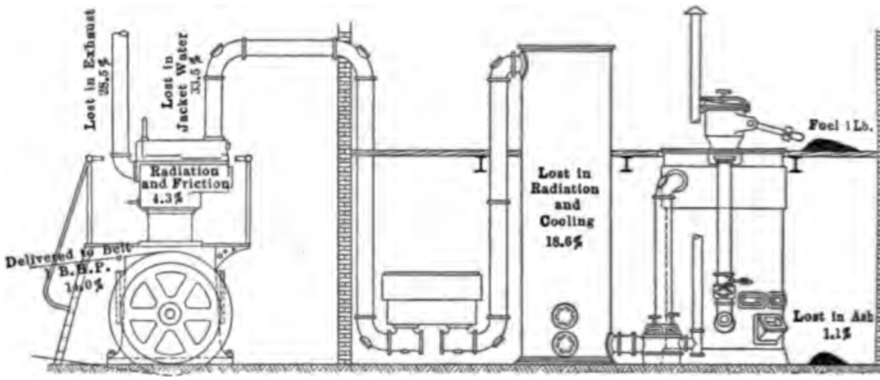


Fig. 256.—Heat distribution—producer-gas plant.

Distribution of Heat

Original heat in the coal = 13,500 heat units per pound

	Steam plant	Gas plant
Heat lost in ashes	2.0 per cent.	1.1 per cent.
Heat lost in radiation and cooling	4.6	18.6
Heat lost in smoke	24.6	.
	——	——
Total losses in boiler and producer	31.2 per cent.	19.7 per cent.
Heat lost in radiation and friction	3.3 per cent.	4.3 per cent.
Heat lost in exhaust	53.5	28.5
Heat lost in jacket water	....	33.5
Heat lost in auxiliaries	7.3	.
	——	——
Total heat losses in entire plant	95.3 per cent.	86.0 per cent.
Net efficiency of plant	4.7 per cent.	14.0 per cent.

Relative Cost of Fuel with Different Types of Installations.—The following tables is, of course, only approximate but it represents average practice, and not the best test results.

Type of plant and fuel	Fuel per b.hp.-hr., pounds	B.t.u. in fuel per b.hp.-hr.	Therm. effi., per cent.	Price of fuel per ton, dollars	Fuel cost, b.hp.	
					Hour, cents	Year,[1] dollars
Small steam plant						
Anthracite coal	4.00	56,000	4.5	2.50	0.50	15.40
Anthracite coal	4.00	56,000	4.5	6.25	1.25	38.50
Anthracite coal	7.00	98,000	2.6	2.50	0.88	27.00
Anthracite coal	7.00	98,000	2.6	6.25	2.19	67.50
Bituminous coal	4.00	52,000	4.9	2.00	0.40	12.60
Bituminous coal	4.00	52,000	4.9	3.50	0.70	21.53
Bituminous coal	7.00	91,000	2.8	2.00	0.70	21.55
Bituminous coal	7.00	91,000	2.8	3.50	1.23	37.80
Large steam plant						
Anthracite coal	2.00	25,000	10.2	2.50	0.25	7.70
Anthracite coal	2.00	25,000	10.2	6.25	0.63	19.25
Bituminous coal	1.30	17,000	15.0	2.00	0.13	4.00
Bituminous coal	1.30	17,000	15.0	3.50	0.23	7.00
Bituminous coal	2.00	26,000	9.8	2.00	0.20	6.16
Bituminous coal	2.00	26,000	9.8	3.50	0.35	10.80
Small producer-gas plant						
Charcoal	1.00	12,000	21.2	12.00	0.60	18.50
Charcoal	1.00	12,000	21.2	20.00	1.00	30.80
Charcoal	1.50	18,000	14.1	12.00	0.90	27.70
Charcoal	1.50	18,000	14.1	20.00	1.50	46.20
Anthracite coal	1.25	15,500	16.4	2.50	0.16	4.80
Anthracite coal	2.00	25,000	10.2	2.50	0.25	7.70
Anthracite coal	1.25	15,000	17.0	6.25	0.39	12.00
Anthracite coal	2.00	25,000	10.2	6.25	0.63	19.25
Coke	1.50	16,500	15.4	3.00	0.23	6.95
Coke	1.50	16,500	15.4	3.00	0.30	11.55
Bituminous coal	1.25	16,500	15.4	2.00	0.13	3.85
Bituminous coal	1.25	16,500	15.4	3.50	0.22	6.78
Bituminous coal	2.00	26,000	9.8	2.00	0.20	6.16
Bituminous coal	2.00	26,000	9.8	3.50	0.35	10.80
Large producer-gas plant						
Bituminous coal	0.80	11,600	22.0	2.00	0.08	2.40
Bituminous coal	1.00	13,000	19.6	2.00	0.10	3.00
Bituminous coal	1.00	13,000	19.6	3.50	0.18	5.40
Bituminous coal	1.50	19,500	13.0	2.00	0.15	4.50
Bituminous coal	1.50	19,500	13.0	3.50	0.26	8.10
Bituminous coal	2.00	21,000	12.1	0.65	0.67	2.00

[1] 3,080 hr.

Type of plant and fuel	Fuel per b.hp.-hr.	B.t.u., b.hp.-hr.	Therm. effi., per cent.	Price of fuel, dollars	Fuel cost, b.hp.	
					Hour, cents	Year,[1] dollars
Gas and oil engines	Cu. ft.					
Natural gas	11.25	11,000	23.1	0.10 M	0.123	3.79
Natural gas	11.25	11,000	23.1	0.30 M	0.37	3.79
Illuminating gas	20.00	12,000	21.2	1.50 M	1.50	45.20
Illuminating gas	20.00	12,000	21.2	1.00 M	2.00	61.60
Crude oil	1.00 pt.	18,000	14.1	0.04 gal.	0.30	15.40
Crude oil	1.40 pt.	25,000	10.2	0.04 gal.	0.70	21.55
Gasoline	1.00 pt.	12,200	20.8	0.15 gal.	1.88	58.00
Gasoline	1.00 pt.	12,200	20.8	0.30 gal.	3.75	115.00
Denatured alcohol	1.00 pt.	9,000	28.3	0.40 gal.	5.00	154.00

In connection with the cost of fuel it is interesting to examine the Government tests of small pumping plants in California.

GASOLINE ENGINES AND CENTRIFUGAL PUMPS

Engine horsepower	Discharge, gal. per min.	Useful water, hp.	Total cost of plant, dollars	Cost per useful water hp.-hr.			
				Fixed charges, cents	Gasoline, cents	Attendance and repairs, cents	Total, cents
3	32	0.145	.....	.....	0.166	.....	
11	112	0.850	1,075	2.530	0.075	.....	2.605
11	147	1.65	3,056	1.235	0.029	0.121	1.285
11	148	2.00	1,200	0.200	0.032	0.042	2.74
15	224	2.46	2,000	0.387	0.024	.....	0.411
21	359	5.63	3,500	0.257	0.019	0.002	0.278
25	635	7.65	2,300	0.036	0.018	.....	0.054
30	399	8.06	3,500	0.080	0.016	0.012	0.108
30	608	10.43	3,343	0.136	0.020	0.001	0.157
35	592	13.51	3,000	0.044	0.016	0.009	0.069

Incidentals, Etc., for These Pumping Engines.—The repair bills, etc., reported for these engines are very small, as indicated below. For one plant consisting of a 23-hp. gasoline engine, single centrifugal pump, a 10-in. well 385 ft. deep, supplying 458 gal. per minute under a 58-ft. head, the total cost of engine and pump, entirely installed with belt and ready to run was $1,382. (This does not include well, pit, building, etc.)

The reported repairs, etc., are:

Incidentals, oil and repairs for 1900	$36.15	= 2.61 per cent. of cost
Incidentals, oil and repairs for 1901	13.50	= 1.02 per cent. of cost
Incidentals, oil and repairs for 1902	23.30	= 1.69 per cent. of cost
Incidentals, oil and repairs for 1903	54.78	= 3.97 per cent. of cost
Incidentals, oil and repairs for 1904	24.65	= 1.78 per cent. of cost
Average	$30.00	= 2.21 per cent. of cost

[1] 3,080 hr.

Examples of Comparative Cost of Operating Different Types of Power Installations.—

Example 1.—The figures presented represent one solution for the probable cost of installing and operating producer gas plants in place of gasoline engines, which operate on distillate, for irrigation purpose in Arizona.

The estimates are for plants of 25, 35 and 50 hp. In the table of estimates the plants operating on distillate are designated "D." Those operating on producer gas by "P-G."

Engine Load.—Seventy-five per cent. normal rating in each case.

Service.—Ten hours per day, 25 days per month.

Water.—As the installations considered are for irrigating purposes, it is assumed that no charge is necessary for circulating water.

The figures given for the quantity of water used are on the assumption that this water is allowed to run to waste.

Labor.—It is assumed that gas-producer operators are not as readily found as operators of gasoline engines and that a higher price must be paid for the former. The wages are $75 and $60 per month.

The entire time of these operators is not required provided they can be near the plants, but in the cases presented it is assumed that the plants are isolated and that the operator has little or nothing else to do. His entire time is, therefore, charged to the cost of operating. Only one operator is required for plants of the size mentioned.

Estimate based on consumption of 1.00 pt. of distillate and 1.25 lb. of anthracite coal per brake horsepower-hour.

Size of plant	25 b.hp.		35 b.hp.		50 b.hp.	
Type of plant	D	P-G	D	P-G	D	P-G
Cost of plant, installed	$1,050	$2,120	$1,450	$2,800	$2,025	$3,750
	Estimated total operating cost per month					
Fuel, gal. and lb.	590	5,850	825	8,250	1,170	11,750
Gal. of lubricating oil	8	8	10	10	12	12
Lb. of waste	10	10	10	10	10	10
Gal. of circulating water	23,500	47,000	31,700	63,500	47,000	94,000
Total cost of fuel	$106.10	$40.95	$148.50	$57.75	$211.00	$82.25
Total cost of lubricating oil	3.20	3.20	4.00	4.00	4.80	4.80
Total cost of waste	1.00	1.00	1.00	1.00	1.00	1.00
Total cost of labor	60.00	75.00	60.00	75.00	60.00	75.00
Operating cost only	$170.30	$120.15	$213.50	$137.75	$276.80	$163.05
Interest on investment, 5 per cent	4.38	8.85	6.05	11.70	8.45	15.60
Depreciation, 8 per cent	7.00	14.10	9.65	18.70	13.50	15.00
Insurance, repairs, taxes, etc., 5 per cent	4.38	8.85	6.05	11.70	8.45	15.60
Total operating cost and fixed charges	$186.06	$151.95	$235.25	$179.85	$307.20	$219.25
Total b.hp.-hr.	4,690	4,690	6,580	6,580	9,370	9,370
Total cost per b.hp.-hr., cents	3.97	3.24	3.57	2.73	3.28	2.34
Total saving per month by installing producer-gas plant	34.11		55.40		87.95	
Difference in first cost	1,070.00		1,350.00		1,725.00	
Months required to pay this difference by saving in operating expenses	31.5		24.50		19.50	

Example 2.—The problem involved in this example is to determine the relative cost of investment and operating cost for a gas-engine installation, operating on natural gas, to be capable of delivering 300 hp. to the countershafts in a machine shop. The four types of installation considered are:

A.	350-hp. engine at $35 per hp.	$12,200	
	Erecting, including air and exhaust pipe	350	
	Starting devices	400	
	Foundation, 100 cu. yd. at $7	700	$13,650
	225-kw. generator A.C. at $12.50 per kw.	2,800	
	12.5 kw. exciter	225	
	Switchboard, etc.	300	3,325
	Fifteen 20-hp. motors at $15.50 per hp.	4,650	4,650
			21,625
B.	Two 175-hp. engines at $37.50 per hp.	13,100	
	Erecting, including air and exhaust pipe	600	
	Starting devices	400	
	Foundations, 53 cu. yd. at $7, (each)	750	14,850
	Two 110-kw. generators at $19 per kw.	4,200	
	Two 10-kw. exciters	380	
	Spring coupling	300	
	Switchboard, etc.	400	5,280
	Motors as in "A"		4,650
			24,780
C.	Three 125-hp. engines at $40 per hp.	15,000	
	Erecting, including air and exhaust pipe	800	
	Starting devices	500	
	Foundations, 40 cu. yd. × 3 at $7	540	17,140
	Three 75-kw. generators at $23.40 per kw.	5,250	
	Three 6-kw. exciters	390	
	Two spring couplings	500	
	Switchboard, etc.		6,640
	Motors as in "A"		4,650
			28,430
D.	Fifteen 20-hp. engines erected on purchaser's foundations, at $43.50 per hp.	13,050	
	Foundations, 15 at $50 each	750	
			$13,800

Estimated Operating Cost per Month.—The operating cost of these installations is given for three sets of conditions, the load on the engines being so changed that it modifies the operating costs as shown. "A," "B," "C," and "D" represent the four types of installations indicated on page 496.

Service, 9 hr. per day for 25 days.

900 B.t.u. gas at 30 cts. per 1,000 cubic feet.

Oil at 35 cts. per gallon.

Waste at 7.5 cts. per pound. (The engineer of this plant is exceptionally economical in the use of waste.)

Water at 5 cts. per 1,000 gal., 6, 7, 10 gal. per horsepower-hour.

Labor for "A," "B," "C," $2\frac{1}{4}$ hr. only per day at 35 cts. per hour.

This plant has a chief engineer now who can run the gas engines if installed in the central plant. The only labor that must be added for "A," "B," "C" is enough to look after the motors.

For "D" it will be necessary to have an engineer at $100 and assistant at $60 per month.

	A		B		C		D	
Engine load, b.hp	350		350		350		280	
Per cent. engine rating	100		100		93.5		93.5	
B.t.u. per b.hp.-hr	11,000		11,000		11,000		12,000	
	Amt.	$	Amt.	$	Amt.	$	Amt.	$
Gas, cu. ft	965 M	289.00	965 M	289.00	965 M	289.00	840 M	252.00
Oil, gal	50	17.50	75	26.25	100	35.00	175	61.25
Waste, lb	25	1.88	35	2.52	45	3.38	150	11.25
Water, gal	273 M	23.65	550 M	27.50	550 M	27.50	630 M	31.50
Labor	......	19.70		19.70		19.70		160.00
Operating only	......	261.73	......	365.07	......	374.58	......	516.00
Interest at 4 per cent	......	72.00	......	82.50	......	95.00	......	48.30
Depreciation 10 per cent	......	170.50	......	206.50	......	237.00	......	121.00
Repairs, etc., 3 per cent	......	54.25	......	61.90	......	71.20	......	(6%) 72.50
Fixed charges	......	206.75	......	350.90	......	403.20	......	241.80
Total operating cost	......	658.48	......	715.97	......	777.78	......	757.80

Saving in operating expense per month compared with "D"	99.32	41.83	− 19.98	
Do. per year	1,192.00	502.00	−240.00	
Excess first cost over "D"	7,125.00	10,080.00	16,930.00	
Time to make up diff. by saving in operating expenses	6 years	20.5 years		

	A		B		C		D	
Engine load, b.hp.	250		250		250		195	
Per cent. engine rating	71.5		71.5		2–100		65.0	
B.t.u. per b.hp.-hr.	12,500		12,500		11,000		13,900	
	Amt.	$	Amt.	$	Amt.	$	Amt.	$
Gas, cu. ft.	780 M	234.00	780 M	234.00	688 M	206.00	678 M	203.00
Oil, gal.	40	14.00	60	21.00	70	24.50	130	45.50
Waste, lb.	25	1.88	35	2.62	45	3.38	150	11.25
Water, gal.	338 M	16.90	394 M	19.70	394 M	19.70	438 M	21.90
Labor	......	19.70	......	19.70	......	19.70	......	160.00
Operating only	286.48		297.02		273.28		441.65	
Fixed charges	306.75		350.90		403.20		241.80	
Total operating cost	593.23		747.92		676.48		683.45	
Saving in operating expense per month compared with "D"	90.22		35.53		6.97			
Saving in operating expense per year compared with "D"	1,088.00		426.00		84.00			
Excess of first cost over "D"	7,125.00		10,280.00		13,930.00			
Time to make up difference by saving in operating expenses	6.6 years		24 years					

	A		B		C		D	
Engine load, b.hp.	175		175		175		130	
Per cent. engine rating	50		100		2–70		43.7	
B.t.u. per b.hp-hr.	14,700		11,000		12,800		16,000	
	Amt.	$	Amt.	$	Amt.	$	Amt.	$
Gas, cu. ft.	644 M	193.00	481 M	144.00	561 M	168.00	525 M	157.50
Oil, gal.	35	12.25	45	15.75	55	19.20	100	35.00
Waste, lb.	25	1.88	35	2.62	45	3.38	150	11.25
Water, gal.	236 M	11.80	236 M	11.80	276 M	13.80	295 M	14.75
Labor	......	19.70	......	19.70	......	19.70	......	160.00
Operating only	238.63		193.87		224.08		378.50	
Fixed charges	306.75		350.90		403.20		241.80	
Total operating cost	545.38		344.77		627.28		620.50	
Saving in operating expense per month compared with "D"	74.92		75.53		3.02			
Saving in operating expense per year compared with "D"	899.00		906.00		36.00			
Time to make up difference by saving in generating expenses	7.9 years		11.4 years					

Example No. 3.—Producer-gas and steam installations.

Size of plant	600 hp.		6,000 hp.	
Type of plant	Gas	Steam	Gas	Steam
Cost of plant	$48,000	$40,000	$420,000	$420,000

Estimated operating cost

....	24 hr. × 300 days		24 hr. × 365 days	
ı engine..................	75 per cent. for 10 hr. 33.3 per cent. for 14 hr.		 75 per cent.	
coal....................	1,300	3,000	21,000	42,000
	$2.75 gross ton		$2.50 gross ton	
of oil....................	1,200 at 30 cts.		10,950 at 30 cts.	
of waste.................	2,000 at 7 cts.		5,000 at 7 cts.	
room labor..............	2 eng. at $17 wk.		1 ch. eng., $100 mo. 3 asst. eng., $17 wk. 3 oilers, $12 wk.	
ıse and boiler room.......	2 men at $15 wk.		2 men at $50 mo.	
....			4 men at $45 mo.	
leaning..................	$4.50 wk.			
water (5 cts. per M)......	18,000 M	125,000 M	246,000 M	1,500,000 M
wasted = 20 per cent.....	3,600 M	35,000 M	49,200 M	300,000 M

Total cost of

....	$ 3,630	8,250	52,500	105,000
....	$ 360	360	3,285	3,285
....	$ 140	140	350	350
room labor..............	$ 1,768	1,768	5,720	5,720
ıse and boiler-room labor..	$ 1,560	1,560	3,600	3,600
leaning..................	$ 234			
....	$ 180	1,250	2,460	15,000
ng expenses only..........	$ 7,922	13,328	67,915	132,955
ıarges at 17 per cent......	$ 8,160	6,800	71,400	71,400
›erating cost.............	$16,082	20,128	139,315	204,355
ilowatt-hours.............	1,555,000		26,000,000	
r kilowatt-hour...........	$ 1.04	1.30	0.536	0.780
ıving per year by operating ıcer gas plant............	$ 4,946		65,040	
ıce in first cost...........	$ 8,000			
equired to pay this differ- by saving in operating ses....................	2			

st of Steam and Blast-furnace Gas Power.—To give an approxi- dea of the relative cost of producing electric power in blast-furnace gine and steam-turbine plants in this country, Mr. H. J. Freyn blished the following data from eight steam-turbine stations and last-furnace gas-engine plants.

Comparison of Cost of Producing Electric Power in Steam and Gas Power Plants

All cost figures in cents per kilowatt-hour

No.	Item	8 steam-turbine plants 1 Max.		2 Avg.		8 blast-furnace gas-engine plants 3 Min.	4 Max.		5 Avg.		6 Min.
1	Plant capacity in kw.	126,000		55,000		10,000	50,000		11,600		1,500
2	Use factor, per cent	33.3		25.0	%	10.0	71.5		49.0	%	22.0
3	Labor	0.1730		0.0902	58.1	0.0434	0.0881		0.0550	33.0	0.0302
4	Repairs and maintenance	0.0740		0.0422	27.3	0.0250	0.1282		0.0733	44.0	0.0273
5	Lubricants	0.0096		0.0054	3.5	0.0020	0.0237		0.0125	7.5	0.0054
6	Water	0.0305		0.0143	9.2	0.0020	0.0162		0.0120	7.2	0.0036
7	Miscellaneous	......		0.0029	1.9	......	......		0.0137	8.3	
8	Total net operating		%					%			
	expense	0.2414	33.3	0.1550	100.0	0.0850	0.2438	52.2	0.1665	100.0	0.0624
9	Cost of 1 million B.t.u., cents	11.10	%	8.80		5.20	10.37	%	8.11		5.80
10	Cost of fuel	0.3960	66.7	0.3100		0.1950	0.2441	47.8	0.1530		0.0062
11	Total cost of power production without fixed		%					%			
	charges		100.0	0.4650				100.0	0.3195		
12	Heat consumption per kw.-hr., B.t.u.	46,400		35,400		29,700	26,000		18,400		16,200
13	Thermal efficiency, per cent	11.49		9.65		7.35	21.0		18.54		13.12
14	Ratio of plant capacities	1		1		1	0.397		0.212		0.150
15	Ratio of use factors	1		1		1	2.130		1.960		2.20
16	Ratio of net operating expenses	1		1		1	1.010		1.074		0.970
17	Ratio of cost of 1 million B.t.u.	1		1		1	0.934		0.922		1.133
18	Ratio of fuel expenses	1		1		1	0.617		0.493		0.494
19	Ratio of actual fuel cost	1		1		1	0.560		0.536		0.436
20	Ratio of total cost of production	1		1		1	.....		0.678		
21	Ratio of heat consumption	1		1		1	0.560		0.520		0.545
22	Ratio of thermal efficiency	1		1		1	1.830		1,923		1,785

Mr. Freyn also published the following figures of "actual total" operating cost of a 40,000-kw. blast-furnace gas-engine plant and of a 49,000-kw. steam-turbine plant for which the use factor happens to be exactly the same.

All items are directly comparable since the fuel cost for the steam-turbine plant was corrected for coal of 10,500 B.t.u. per pound at $1.80 per long ton, which is the basis for valuation of blast-furnace gas in this gas-engine plant.

COMPARISON OF COST OF PRODUCING ELECTRIC POWER

All cost figures in cents per kilowatt-hour

	Gas-engine plant				Steam-turbine plant			
Year	1910				1910			
Capacity in kw.	40,000				49,000			
Kw.-hr. produced	116,535,000				142,835,000			
Use factor, per cent	33.3				33.3			
Net operating expenses:		%				%		
Labor	0.0678	52.0			0.0528	52.0		
Repairs and maintenance	0.0366	28.0			0.0326	32.0		
Lubricants	0.0116	9.0			0.0024	2.5		
Water	0.0074	6.0			0.0073	7.0		
Miscellaneous	0.0064	5.0		%	0.0066	6.5		%
Total	0.1298	100.0	0.1298	40.0	0.1017	100.0	0.1017	23.0
Cost of 1 million B.t.u.	9.5[1]				9.9			
Fuel cost			0.1951	60.0			0.3400	77.0
Total cost of power production (without fixed charges)			0.3249	100.0			0.4417	100.0
Heat consumption B.t.u. per kw.-hr.	19,500[1]				35,200			
Thermal efficiency, per cent	17.5[1]				9.7			

Actual Fuel Consumption and Cost of Operation of Existing Plants.—The following tables and diagrams were taken from Neuere Kraffanlangen by Prof. Josse and were calculated from data secured by the author from manufacturing plants, plants of business houses and electrical stations—principally the latter. They show what may be regarded as average results of German practice in 1911.

The results for the individual plants differ materially due to the construction of the engine plant, its maintenance, operation and the number of employees. Interest and amortization expense are not considered because these depend upon local conditions.

In Table 1 are given, for reciprocating engine plants of capacity less than 1,000 kw., the maximum continuous load, the plant cost, yearly load, thermal efficiency and the cost of fuel, lubricants, packings and supplies, maintenance and total cost.

Table 2 gives a similar tabulation in so far as data are available for locomobiles

Table 3 gives a similar tabulation for two small turbine plants.

Steam-turbine plants of small capacities are still hard to find. The thermal efficiency is good (0.08–0.09) with a total cost of 1.56 cts. per kilowatt-hour. The reason is due more to the fact that the turbines are in modern plants and require less maintenance than to any thermal superiority of the turbine.

[1] Approximate.

TABLE 1.—RESULTS OF AN AVERAGE YEAR

Reciprocating steam-engine plants (up to 1,000-kw. capacity)

Plant equipment	Maximum continuous load, kw.	Plant cost of mechanical and electrical machinery per kw.,[1] dollars, kw.	Yearly load, kw.-hr.	Thermal eff.	Operating cost (cents per kw.-hr.)				
					1 Fuel	2 Lubricants, packings and supplies	3 Salaries and wages	4 Maintenance	5 Total operating cost (1–4), etc.
2 boilers, 3 compound engines	78	254.50	120,556	0.040	1.860	0.214	0.822	0.242	3.88
2 boilers, 3 single-cyl. engines	87	204.50	108,428	0.041	1.780	0.133	0.655	0.329	3.40
2 boilers, 3 compound engines	160	141.50	238,648	.....	1.300	0.193	0.784	0.100	3 40
2 boilers, 2 compound engines	170	......	152,860	.....	1.395	0.126	1.130	0.357	3.91
3 boilers, 2 compound engines	264	152.00	901,390	0.082	0.640	0.033	0.636	0.312	1.81
2 boilers, 2 compound engines	302	141.50	224,180	0.069	0.920	0.121	1.070	0.492	2.82
3 boilers, 3 compound engines	310	82.50	331,113	0.070	0.970	0.062	0.775	0.284	2.26
3 boilers, 3 compound engines	340	139.00	610,810	0.048	0.810	0.072	0.535	0.286	1.80
3 boilers, 4 compound engines	363	184.50	513,298	0.058	1.365	0.068	0.497	0.147	2.50
3 boilers, 2 compound engines	400	127.00	537,516	0.071	0.762	0.195	1.153	0.355	2.69
3 boilers, 3 compound engines	400	150.00	719,630	0.053	0.905	0.053	0.358	0.191	2.24
2 boilers, 2 compound engines	400	141.50	833,446	0.086	0.643	0.083	0.676	0.257	1.83
3 boilers, 3 compound engines	405	147.00	697,236	0.070	0.750	0.038	0.750	0.357	2.02
2 boilers, 3 compound engines	416	130.50	534,700	0.055	0.740	0.031	0.955	0.620	2.40
6 boilers, 3 compound engines	420	131.00	332,037	0.062	1.120	0.071	0.548	0.348	2.10
3 boilers, 3 compound engines	450	......	440,805	0.049	1.760	0.078	0.703	0.381	3.10
4 boilers, 5 compound engines	471	113.00	581,867	0.061	0.823	0.068	0.960	0.216	2.60
3 boilers, 2 comp. eng., 1 turbine	500	111.50	247,300	0.069	0.835	0.081	1.370	0.453	2.95
2 boilers, 2 comp. eng	600	140.00	688,100	0.060	1.010	0.159	0.428	0.188	2.36
4 boilers, 3 comp. eng	880	102.50	1,019,863	0.069	0.990	0.064	0.724	0.183	2.25

[1] Engines, boiler and setting, feed-water pump, condensers, cooling apparatus pipes, transformers exclusive of cables and accumulators.

TABLE 2.—RESULTS OF AN AVERAGE YEAR

Locomobiles

Plant equipment	Maximum continuous load, kw.	Plant cost of mechanical and electrical machinery per kw., dollars, kw.	Yearly load, kw.-hr.	Thermal eff.	Operating cost (cents per kw.-hr.)				
					1 Fuel	2 Lubricants, packings and supplies	3 Salaries and wages	4 Maintenance	5 Total cost (1–4), etc.
2 locomobiles	65	95.30	51,106	0.045	1.356	0.071	0.975	.....	3.34
2 locomobiles	70	85.30	58,000	.....	1.142	0.064	1.070	0.033	3.26
2 superheat locomobiles at 80 kw	160	.....	107,520	.....	0.744	0.181	0.465	0.328	1.71
2 superheat locomobiles at 90 kw	180	.....	131,487	.....	1.015	0.214	0.460	0.124	1.81
2 compound superheat locoombiles at 100 kw.	200	.....	316,442	0.074	0.937	0.147	0.295	0.097	1.48
3 compound superheat locomobiles at 100 kw.	350	97.20	300,249	0.076	0.738	0.045	0.530	0.007	1.38
4 compound superheat locomobiles at 100 kw.	356	.....	832,270	.....	0.905	0.064	0.410	0.132	2.81

TABLE 3.—RESULTS OF AN AVERAGE YEAR

Steam-turbine plants of smaller capacities (up to 1,000 kw.)

2 turbines at 200 kw	400	103.5	289,852	0.08 (1908) 0.09	1.000	0.012	0.762	.....	1.81
3 turbines, 2 at 200 kw., 1 at 300 kw	700	.....	1,250,000	.....	.....	.....	...	.....	1.56

TABLE 4.—RESULTS OF AN AVERAGE YEAR

Large steam power plants (steam engines and turbines) over 1,000 kw. continuous load

Plant equipment	Maximum continuous load, kw.	Plant cost of mechanical and electrical machinery per kw., dollars, kw.	Yearly load, kw.-hr.	Thermal eff.	Operating cost (cents per kw.-hr.) 1 Fuel	2 Lubricants, packings and supplies	3 Salaries and wages	4 Maintenance	5 Total operating cost (1–4), etc.
5 boilers, 4 recp. engs. and turbs.	1,100	113.80	1,562,400	0.056	0.490	0.084	0.398	0.230	1.38
4 boilers, 4 recp. engs. and turbs.	1,146	125.00	1,473,350	0.073	0.765	0.057	0.643	0.193	2.10
6 boilers, 4 recp. engs. and turbs.	1,180	96.50	2,669,000	0.060	0.662	0.033	0.505	0.250	1.63
5 boilers, 6 recp. engs. and turbs.	1,760	......	1,982,300	0.060	0.704	0.048	0.372	0.133	1.33
6 boilers, 5 recp. engs. and turbs.	2,786	85.30	1,788,200	0.055	0.953	0.064	0.515	0.002	1.84
10 boilers, 7 recp. engs. and turbs.	2,900	121.00	6,111,536	0.062	1.010	0.052	0.372	0.057	1.78
8 boilers, 5 recp. engs. and turbs.	3,535	69.00	3,625,000	0.064	0.690	0.029	0.326	0.150	1.50
20 boilers, 5 recp. engs. and turbs.	3,850	88.00	8,147,848	0.058	0.290	0.026	0.357	0.064	0.92
12 boilers, 4 recp. engs. and turbs.	4,500	138.00	11,475,260	0.115	0.369	0.228	0.334	0.150	0.93
16 boilers, 9 recp. engs. and turbs.	4,680	83.00	7,466,119	0.072	0.940	0.038	0.357	0.061	1.43
11 boilers, 7 recp. engs. and turbs.	5,000	......	9,932,400	0.057	0.775	0.033	0.393	0.072	1.48
22 boilers, 9 recp. engs. and turbs.	8,000	118.00	21,707,286	0.086	0.481	0.038	0.262	.....	1.15
16 boilers, 13 recp. engs. and turbs.	8,710	85.50	10,616,323	0.084	0.465	0.055	0.390	0.212	1.33
14 boilers, 4 recp. engs. and turbs.	9,000	71.80	14,602,890	0.076	0.880	0.040	0.250	0.088	1.31
123 boilers, 46 recp. engs. and turbs.	78,332	......	193,400,547	0.116	0.483	0.017	0.150	0.100	1.06

Table 4 shows results for large steam plants (of over 1,000-kw. capacity). In such plants more reciprocating engines than turbines are found.

Table 5 gives results for steam-turbine plants of more than 1,000-kw. capacity. Lubrication expense is reduced and the cost of fuel and wages determine the cost of operation which in one plant is reduced to 0.525 cts. per kilowatt-hour with a thermal efficiency of 12.3 per cent.

Table 6 gives a tabulation for gas power plants. These costs differ depending on the fuel used. Most plants use anthracite or coke or both, the specific cost being lower with coke. Reliable information was not available of plants using lignite briquettes, but these are somewhat under those for coke. Gas plants are thermally superior to steam plants. The diversity in the specific cost of fuel is partly explained by the difference in price. The cost of fuel decreases as the load increases only up to 300,000 kw.-hr. yearly load, and the expense of fuel is independent of the size of plant.

The fuel cost varies for two similar plants of the same yearly load from 0.38 ct. to 0.88 ct. per kilowatt-hour. The cost of maintenance and the cost of lubrication are about the same as for the steam plant. For small gas plants the maintenance cost is greater than for the steam plant, but for a yearly load of 700,000 kw.-hr. the costs remain below that for steam plants. The total costs vary from 1.5 cts. to 2.66 cts. per kilowatt-hour. At about 600,000 kw.-hr. the total operating cost is about 1½ times the fuel cost. Up to 10 years ago there were no gas engines built of more than 1,000 hp. capacity. Now there are more than 1,000,000 hp. in operation in the world. Two fuels are in common use: Blast-furnace gas and coke-oven gas.

In Table 7 are given the operating results for two large gas-engine plants. This sort of information is difficult to secure as these plants are just beginning to arrange for the exact measurement of gas used. The total cost is low, ranging from 0.264 ct. to 0.357 ct.

In Table 8 are shown Diesel engine results. The thermal efficiency is nearly the same for all the plants (about 31 per cent.) with the exception of the smallest. The cost of lubricants, etc., is somewhat higher than for steam and gas plants. The maintenance can be almost neglected. The unit costs for salaries decrease inversely with the yearly load. In general they require more skilled attendance. The total cost of operation amounts to 1.79 cts. for small plants and falls to 1.095 cts. for a plant with a yearly load of 1,000,000 kw.-hr. The total cost amounts to 1.7 times the fuel cost for plants of large size and about double the fuel cost for small plants.

Fig. 185 shows, in the lower half of the diagram, how the total operating costs and the fuel costs for steam, gas and Diesel engines compare.

TABLE 5.—RESULTS OF AN AVERAGE YEAR

Large steam-turbine plants (of over 1,000 kw. maximum continuous load)

Plant equipment	Maximum continuous load, kw.	Plant cost of mechanical and electrical machinery per kw., dollars, kw.	Yearly load, kw.-hr.	Thermal eff.	Operating cost (cents per kw.-hr.)				
					1 Fuel	2 Lubricants, packings and supplies	3 Salaries and wages	4 Maintenance	5 Total cost, (1–4), etc.
4 turbines	1,167	74.00	1,010,266	.....	0.512	0.023	0.671	0.113	1.430
3 turbines	1,500	54.70	1,257,191	0.062	0.650	0.064	0.541	0.029	1.530
4 turbines	1,850	83.50	3,408,000	0.074	0.475	0.023	0.536	0.057	1.168
3 turbines	3,125	52.40	3,109,210	0.095	0.520	0.007	0.216	0.019	0.835
2 turbines	6,000	61.50	13,325,200	0.106	0.440	0.007	0.131	0.081	0.726
3 turbines	15,000	50.20	25,300,000	0.123	0.338	0.003	0.095	0.031	0.525

TABLE 6.—RESULTS OF AN AVERAGE YEAR

Gas power plants

Plant equipment	Max. continuous load, kw.	Plant cost, mechanical and electrical machinery per kw., dollars, kw.	Yearly load, kw.-hr.	Thermal eff.	Operating cost (cents per kw.-hr.) 1 Fuel	2 Lubricants, packings and supplies	3 Salaries and wages	4 Maintenance	5 Total cost (1–4), etc.	Fuel used
2 engines, 2 generators	32	......	65,850	.....	0.914	0.0953	1.330	0.334	2.665	Anthracite.
2 engines, 2 generators	50	......	114,000	.....	0.505	0.0715	1.140	0.238	1.955	Brown coal briquettes.
2 illuminating gas	67	......	88,069	.....	1.528	0.0524	1.020	.....	4.190	Illuminating gas.
1 engine, 1 generator	100	......	372,000	.....	0.300	1.1430	0.763	0.143	1.420	Brown coal briquettes.
2 engines	108	175.00	156,086	.....	0.847	0.1190	1.080	0.612	2.660	Illuminating gas.
2 engines, 2 coke generators	144	196.00	206,141	0.188	0.370	0.0524	0.965	0.221	2.200	
2	200	151.00	376,314	0.148	0.415	0.0500	0.740	0.202	1.500	Coke.
3	230	......	335,763	0.104	1.000	0.1000	0.785	0.570	2.540	Anthracite.
3 engines, 2 pressure generators	234	183.00	463,326	0.116	0.657	0.0690	0.630	0.240	1.915	Anthracite coke.
3 engines, 3 pressure generators	252	197.50	312,760	.....	0.680	0.0620	0.858	0.240	1.860	Anthracite coke.
3 engines, 2 pressure generators	260	162.00	277,326	0.108	0.885	0.1355	0.535	0.091	2.380	Anthracite coke.
2 double-acting engines, 3 coke generators	270	139.00	275,310	0.091	0.383	0.0870	0.575	0.102	1.528	
3	332	176.00	494,673	0.142	0.488	0.0262	0.762	0.193	1.775	
2 single-acting	400	99.00	426,643	0.135	0.755	0.0905	0.745	0.266	2.000	Anthracite.
4 four	470	130.00	693,810	{ 0.073 0.212 Ill.	0.615	0.0620	0.240	0.243	1.580	Illuminating gas, brown coal briquettes and coke.
4	500	145.00	762,305	.....	0.416	0.0833	0.820	0.086	1.595	Coke.
4	570	106.00	1,222,016	0.186	0.953	0.0761	0.452	0.330	2.520	Illuminating gas.
6	1,400	195.00	3,004,672	.....	0.585	0.0428	0.525	0.095	1.760	Illuminating gas.

TABLE 7.—RESULTS OF AN AVERAGE YEAR

Large gas-engine plants

Plant equipment	Hp.	Fuel	Thermal eff.	Operating cost (cents per kw.-hr.) Fuel	Salaries and wages	Lubricants, packings and supplies	Repairs and gas cleaning	Total
6 gas blast engine sets	6,400							
1 gas blast engine set	2,000	Blast-furnace gas					•	
1 gas blast engine set	2,000							
6 gas engine dynamo sets	5,200	Heating value = 100 B.t.u. per cubic foot.	0.214	0.1605	0.0608	0.0231	0.0197	0.264[1]
1 gas engine dynamo set	1,100							
2 gas engine dynamo sets	800							
	17,500							
3 gas engines	600							
2 gas engines	600	Blast-furnace gas						
1 gas engine	600							
2 gas engines	2,800	Heating value = 100 B.t.u. per cubic foot.	.....	0.1430	0.0525	0.0858	0.0762	0.357
2 gas engines	4,000							
1 gas engine	2,500							
	11,100							

[1] Not considering, water, special repair parts, power necessary for water supply, gas-cleaning, general expenses, interest and depreciation expense.

TABLE 8.—RESULTS OF AN AVERAGE YEAR

Diesel engine plants

Plant equipment	Maximum continuous load	Plant cost of mechanical equipment per kw., dollars, kw.	Yearly load	Thermal eff. = eff. hp. in heat units, heat in fuel	Operating cost (cents per kw.-hr.)				
					1 Fuel	2 Lubricants, packing and supplies	3 Salaries and wages	4 Maintenance	5 Total cost (1–4), etc.
2 engines, 50 hp. each	(a) 75	......	97,567	0.266	0.953	0.143	0.596	.....	1.790
2 engines, 100 hp. each	(a) 148	165.50	204,197	0.306	0.780	0.107	0.845	0.107	1.833
2 engines, 100 hp. and 200 hp.	(a) 220	......	....	0.310	0.730	0.071	.....	.....	
2 Diesel engines	(a) 264	166.50	347,954	0.250	0.685	0.038	0.620	0.021	2.020
2 engines	(a) 330	138.00	....	.....	0.628	0.136	1.070	0.262	2.840
2 Diesel engines	(a) 330	124.00	181,907	0.320	0.543	0.162	0.338	0.083	1.120
3 engines	(a) 440	......	389,787	0.313	0.725	0.081	0.610	.....	1.410
3 4-cyl. engines, 300 hp. each	(a) 664	......	627,324	0.318	0.736	0.186	0.455	.....	1.380
3 cyl. engines, 300 hp. each	(a) 664	130.00	1,037,420	0.323	0.704	0.133	0.252	.....	1.905
	1,600 hp.	......	2,694,500	0.305	0.243	.....	.....	.....	
	1,600 hp.	......	7,440,000	.....	0.341	0.033	0.098	.....	0.476 Russia.

(a) Approximate.

In the upper portion the cost for wages, maintenance, lubricants, etc., are expressed in percentage to the total cost for each of these types of engines. The gas engine shows the smallest fuel expense; the Diesel engine the smallest total operating expense, the steam-engine plant being in both cases the most expensive. In the upper chart, the Diesel engine is the lowest and the gas engine the highest.

CHAPTER XXIV

COMPRESSED AIR

Compressed air is used for transmitting power, for the storage of energy for many purposes, and for producing refrigeration. Air at moderate pressures is used in blast-furnace work and in the Bessemer process; air at higher pressures for the transmission of power, the operation of cranes, hoists and presses, and for the working of motors such as drills, coal-cutting machinery, hoists, street cars and similar applications. High-pressure air has been used for storage, for refrigeration and in certain chemical processes. Air at low pressures, between atmospheric and 5 or 6 lb. per square inch, produced by centrifugal fan blowers or the so-called positive blowers, is used for the ventilation of mines, buildings and ships and for producing forced and induced draft for steam boilers. The storage of energy by compressed air usually differs from the transmission of power, in that the compressed air, which is forced into the receiver at high pressure, is generally used at a much lower pressure in the air motor.

Although compressed air has been used in engineering operations for a period of probably 200 years, the modern application of compressed air is probably due to Messrs. Kraft and Sommeiller whose extensive experiments at the Cockerill works at Seraing in Belgium, resulted in the use of compressed air at the Marie colliery in Seraing in 1856. The Sommeiller compressor built by the Cockerill Co. compressed the air for the work in the Mount Cenis tunnel in 1861. The air pressure used was 106 lb. per square inch and the longest transmission lines exceeded 20,000 ft. The air motors worked expansively, the cylinders being heated to prevent freezing. The Sommeiller compressor (see Fig. 257) consisted of a plunger working between two containers filled with water, the water serving to cool the air and acting as the piston of the compressor. These compressors were quite economical and more modern constructions on the same principle, such as the Leavitt compressor at the Calumet and Hecla copper mines, have given very good service. This compressor has double-acting cylinders, 60 by 42 in. and runs at the comparatively high speed of 25 r.p.m. About 1868 the dry compressor came into use, the cooling being imperfect. This was improved shortly afterward by the introduction of the spray injection by Prof. Colladon. Spray injection is now no longer used and both cylinders and pistons are water-cooled to reduce the loss by heating as far as possible. It was soon found that it was much better to make the compressor a compound or two-stage machine and to install intercooling coils between the cylinders. Modern air compressing dates from about 1877, when Mekarski and Popp commenced the installation of compressed-air plants for the driving of railways and the distribution of power. In the United States the development of compressed air has followed the development of the mining industry, and most of the compressors have been built and sold for the working of air drills and similar machinery in the mining and quarrying fields. About 15 years ago the "pneumatic tool" came into the market and since

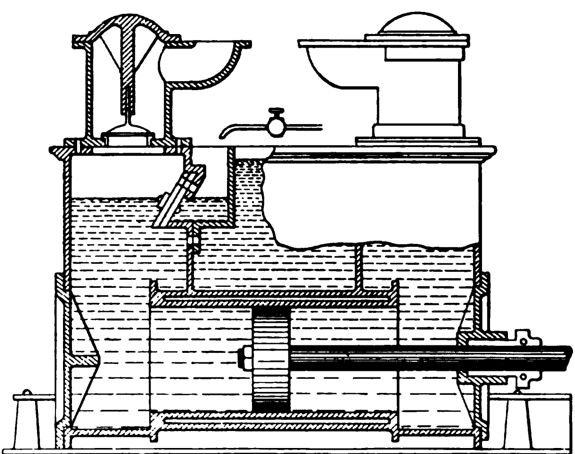


FIG. 257.—Sommeiller hydraulic compressor.

that time no shop or manufactory is complete without its compressed-air line, which supplies power for the use of an infinite variety of tools. Kraft in the years 1854–8 used compressed air for the cranes at the Cockerill works at Seraing and many of these cranes are still in use. Pneumatic lifts and cranes are now installed in many places.

Air-compression machinery may be divided into (*a*) piston compressors and blowing engines; (*b*) rotary blowers of the positive-pressure type; (*c*) centrifugal blowers or fans including the turbo-blower and compressor; and (*d*) the hydraulic air compressor.

Piston Compressors and Blowing Engines.—Piston compressors and blowing engines differ only in the pressure to which they work. Blowing engines for the blast furnace usually work from 8 to 15 lb. per square inch above the atmosphere. Blowing engines for the Bessemer converter work between 18 and 35 lb. gage. Ordinary air compressors of the piston type for power purposes are usually built to deliver air between 50 and 80 lb. per square inch for single-cylinder compression. From about 70 to 120 lb. per square inch they are usually compounded and for higher pressures, up to 2,000 to 2,500 lb. per square inch, three- and four-stage compression is used with intercoolers between each two stages. The construction is practically that of a steam engine, Fig. 269, the only differences being the unusual care taken with the jacketing and intercoolers, the excessively small clearance and the type, location and area of the valves. The early air compressors used the standard steam-engine valves with the consequence that the volumetric efficiencies were in many cases below 50 per cent. This almost immediately led to the use of poppet discharge valves and very large mechanically moved inlet valves, and finally to mechanically controlled valves of very large area for both suction and discharge. The well-known Reidler valves, Fig. 258, were invented for this use. These valves, however, have been superseded on the more modern machines by a very light spring-controlled multiported diaphragm valve, known as the Borsig type, Fig. 259, which is now used by practically all of the better class of builders on low- and medium-pressure work. The International Pump Co. make a straight leaf valve of this type. For the large blowing tubs of blast-furnace and steel-mill work the Slick system has been very largely used. Here the cylinder heads are firmly fixed to the base of the machine, while the cylinder barrel is provided with slotted suction openings on each end and is moved backward and forward at the proper time, to ensure full valve opening as in Fig. 261. The discharge valves are located as usual in the heads. Some of these blowing tubs have been built as large as 100 in. in diameter with a stroke of 7 ft. Bessemer blowing engines are usually of much smaller size and are built either on the Slick system or with the Borsig valves (see Trinks paper, vol. 33, *Transactions* A.S.M.E.).

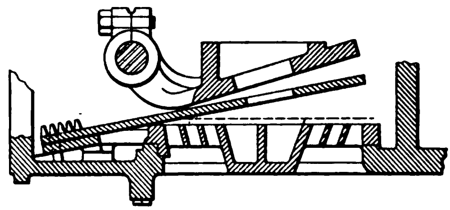


FIG. 258.—Riedler valve.

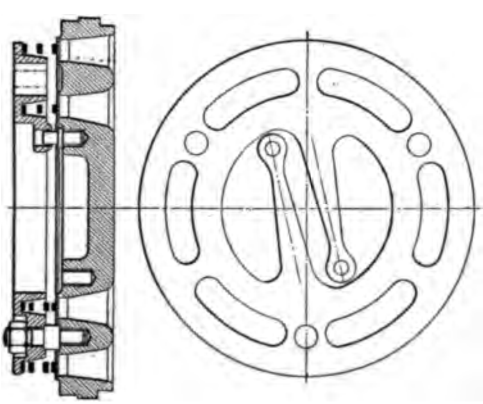


FIG. 259.—Borsig valve for blowing engines or air compressors.

The positive pressure blower, Fig. 262, consists of two shafts carrying two- or three-lobed impellers running in a casing with extremely small clearances. These blowers may be used up to about 15 lb. per square inch and are built with much success in this country by such firms as the Connersville Blower Co. and the P. & F. M. Root Co. These blowers

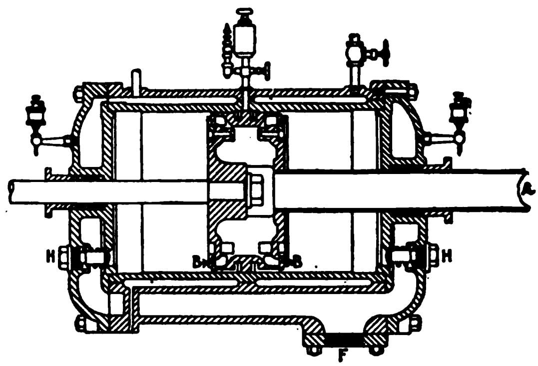


FIG. 260.—Compressor cylinder with piston intake.

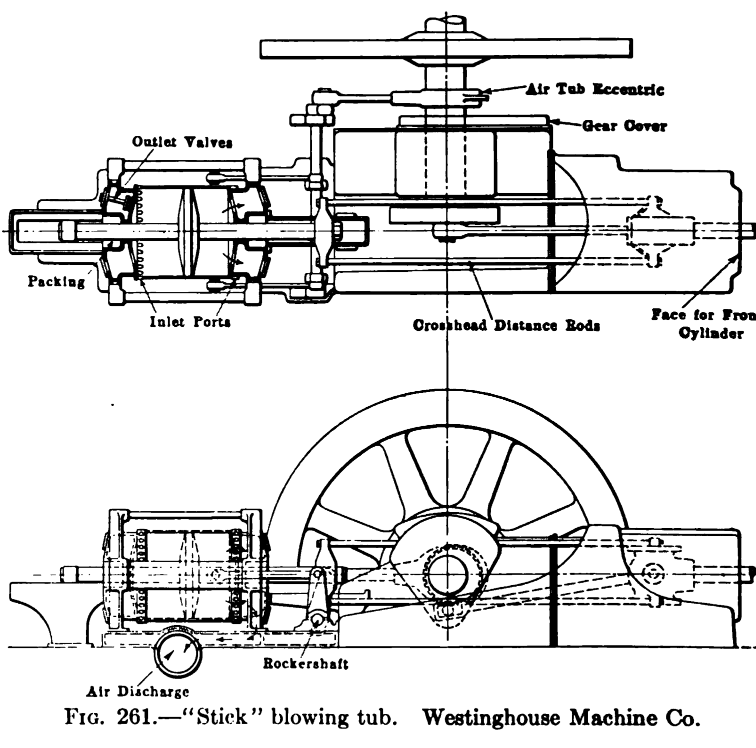


FIG. 261.—"Stick" blowing tub. Westinghouse Machine Co.

may be used either as blowers or exhaustors or as pumps (see *Transactions* A.S.M.E., vol. 24, paper by J. T. Wilkin; vol. 28, paper by Gregory).

The centrifugal blowers are of two varieties, the volume blower, used exclusively for low pressures and large volumes, up to say 15 or 20 in. of water, and the so-called high-pressure blower or cupola blower, which delivers a smaller volume at pressures not exceeding 2 or 3 lb. per square inch. Centrifugal compressors of the "turbo" variety with many stages, Fig. 263, may be built for use up to a pressure of about 150 lb. per square inch.

Turbo-blowers and compressors are similar machines, differing only in the delivery pressures and the number of stages and delivery volume. They are usually built on the principle of the centrifugal pump, that is, the fluid to be compressed is conducted in radial paths, the only design departing from this arrangement is the Parsons, where the flow direction is axial. Turbo-blowers for blast-furnace work will deliver from 17,000 cu. ft. per minute to 60,000 per minute. Bessemer blowers deliver up to about 30,000 cu. ft. per minute. Turbo-compressors have in general a much lower capacity, not usually exceeding 20,000 cu. ft. a minute at from 75 to 135 lb. gage. Larger turbo-compressors have been built, up to a capacity of 50,000 cu. ft. per minute. Steam-turbine drive is usually used where great capacity fluctuations obtain, whereas with constant capacity the high-speed electric motor is most used. With smaller outfits the steam turbine is almost invariably found, as with this drive higher speeds may be obtained and the number of stages kept down. The number of stages varies between one for very low-pressure machines to 28 for the higher pressures. Up to 14 stages compressors are usually built on one shaft and in one casing. With a higher number of stages two casings are used. All of these machines are very carefully water-cooled, but the cooling does not appear to be as efficient in the lower stages as in

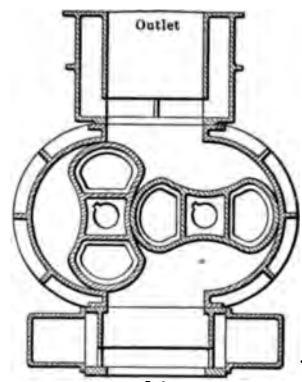


FIG. 262.—Connersville blower or pump.

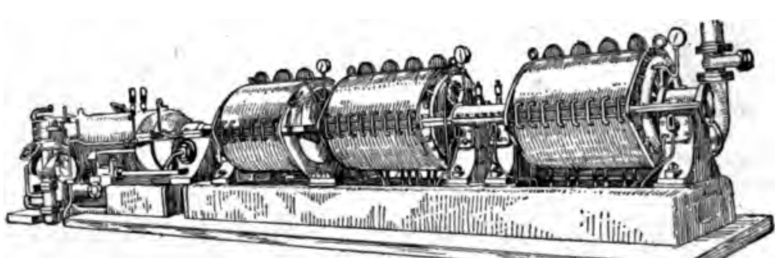


FIG. 263.—Rotary air compressor, turbine driven.

the upper ones. For high-pressure work this is a disadvantage for the turbo-compressor. However, the cooling in the upper stages is so good that the discharge temperature of the air, when working at 115-lb. pressure can be brought down to about 165°F. with water at the ordinary temperatures. A 1,000-hp. compressor, under these conditions, will use about 1,600 cu. ft. of water per hour for cooling. The minimum size of compressors, up to 100 lb., is in the neighborhood of 2,500 cu. ft. per minute, while for blowers at 20 lb., the minimum may be taken at 5,000 cu. ft. per minute. The speeds of revolution run from 5,000 to 6,000 in the smaller sizes down to 3,000 in the larger sizes of machines. The construction of the impellers is usually of radial buckets riveted between two nickel-steel disks, although cast wheels have also been used. Pressures up to 90 lb. have frequently been obtained in as low as 12 stages, but the buckets in this case were not radial. (See paper by Richard S. Rice, *Transactions* A.S.M.E., vol. 33, p. 381, for discussion of the turbine blower with efficiencies and costs.)

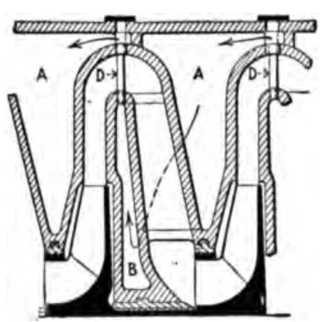


Fig. 264.—Section of turbine air compressor showing water-cooling arrangement.

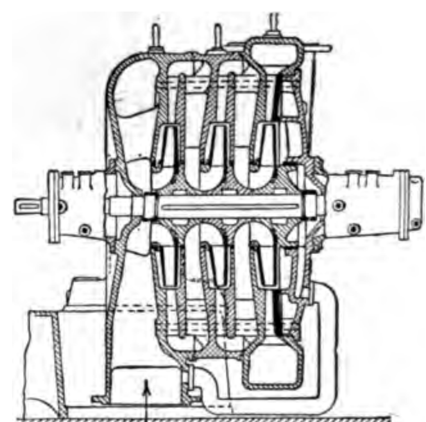


Fig. 265.—Section of three-stage turbine air compressor.

In the hydraulic air compressor, see Fig. 266, a descending column of water is allowed to draw into itself a certain amount of air. At the lowest point of the apparatus a sudden enlargement of section and change of direction slows down the water velocity to such an extent that the entrained air is set free and is collected in a pocket. This air, which is compressed to the pressure due to the head of water at this point, is piped to the surface for distribution and use. Such a plant is installed near Greenville, Conn., on the Quinnebaug River and has been quite successful. The efficiency of the apparatus is very high, but it is large and costly and can only be used in very advantageous locations. For an account of the Greenville plant see Webber's paper, A.S.M.E., vol. 22, page 599. Frizell's paper in the *Journal of the Franklin Institute*, 1880, also contains a test on a plant of this kind, but a full discussion of the subject is in Parker, "The Control of Water."

Compressed-air motors are usually of the type of the steam engine but for small powers a rotary machine of the impulse turbine type has been used. Pulsometer, Emerson and other fluid pressure pumps are worked occasionally by air, especially where there is danger of flooding and occasionally it is convenient to use air in the ordinary duplex pump.

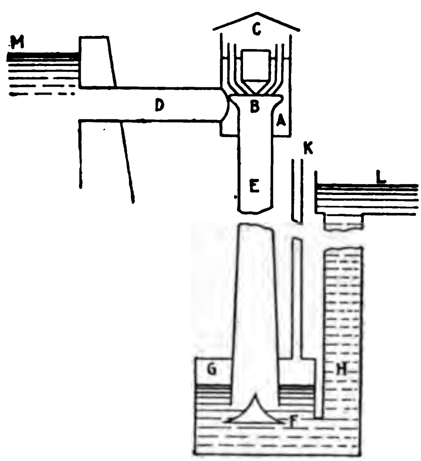


FIG. 266.—Hydraulic air compressor.

Economy in the use of air can be secured by preheating either by steam or outside heat or by a gas flame in the air current. Where preheating is used it may be possible to get more power out of the air than the work of compression and the higher the pressure and degree of preheating the larger is the saving. Where a compound motor is used as in some of the mine hoists both preheating and reheating may be practised with consequent economy. In the Porter compound mining locomotive the air is preheated, used in the high-pressure cylinder and expanded to a low temperature, possibly $-30°$. It is reheated by the heat of the atmosphere blown through the reheater and then used in the low-pressure cylinder.

Air-lift Pump.—The air-lift pump, Fig. 267, is the reverse of the hydraulic air compressor and finds a wide application in pumping deep wells. Compressed air is led through a small pipe to the bottom of the casing and the difference in weight between the water outside the casing and the mixture of air and water inside starts the well flowing. (See A.S.M.E. *Transactions*, vol. 31, page 311 and Parker, "The Control of Water.")

The mechanical efficiency of first-class air compressors, driven by first-class steam or gas engines, is about 85 per cent.; ordinary machines will run below this. The mechanical efficiency of turbo-blowers and compressors will usually run around 90 per cent. The overall efficiency in the best machines will run from 70 per cent. downward.

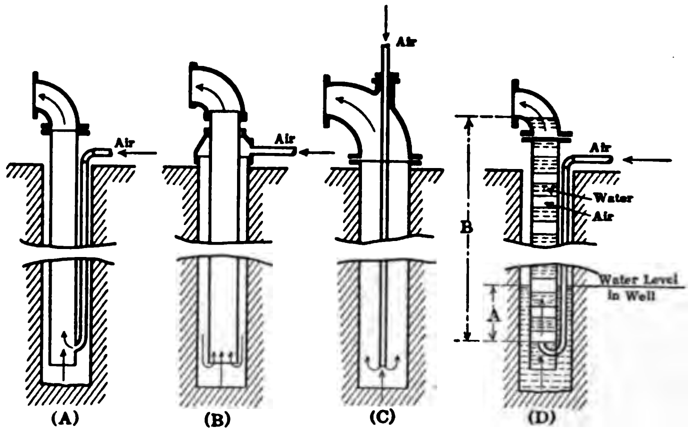


Fig. 267.—Air lifts.

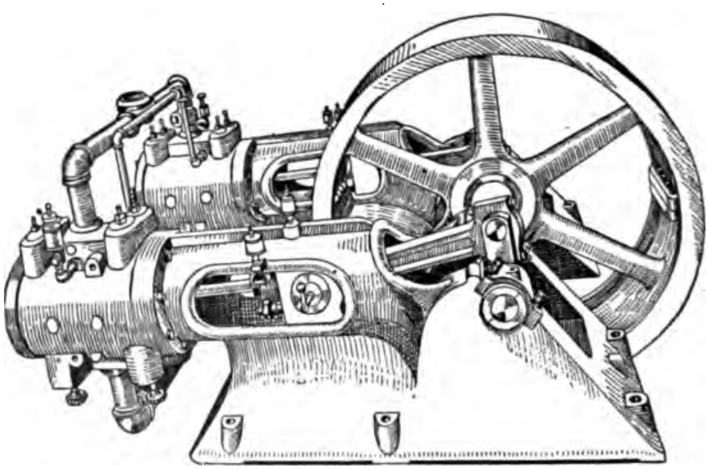


Fig. 268.—Belt-driven duplex compressor.

Volumetric Efficiency.—The actual capacity of compressor cylinders is not equal to the apparent capacity, due to the effect of clearance, heating of the intake air and imperfect valve action. There have been very few tests made to show measured volumetric efficiencies, but where they have been made, as in the case of the Rand mines these efficiencies were shown to be around 60 per cent. Standard machines should give from

85 to 95 per cent. volumetric efficiency under ordinary conditions. It frequently happens that the air ducts bringing the outside air to the intake valves of the compressors are so designed that a considerable rise of temperature takes place within them, together with a loss of pressure. Such arrangements have often resulted in the reduction of volumetric efficiency by as much as 15 to 20 per cent.

Oil.—A number of explosions have taken place in air storage tanks and compressing cylinders due to the vaporization of the lubricating oil used in the air cylinders, hence the greatest care should be taken in the choice of the lubricant, and only the necessary quantity should be used. Some compressors have been lubricated with colloid graphite, and water lubrication has been used with success. Turbo-compressors need no lubrication and will probably be more largely used on this account.

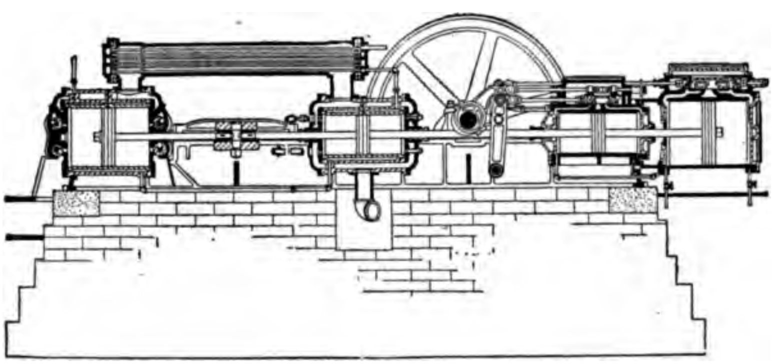


FIG. 269.—Tandem compound steam and two-stage air compressor straight-line type.

The transmission of power by compressed air has been quite attractive in the past, especially where power for compression was cheap and abundant, and although displaced by electric transmission for many purposes, has still a large field for use. As the first extensive experiments were made at Seraing in connection with the mines, so in mining operations compressed-air transmission finds its greatest development today. It is also used for operating cranes and other machines where power is used only at intervals, as the condensation of steam, when used directly, is excessive and hydraulic power is liable to give trouble from freezing. The first large system installed for actual commercial work was at Paris, where Popp in 1881 built the St. Fargeau station (2,000 hp.) and later the station at the Quai de la Gare (10,000 hp.). The system had 34 miles of air mains, including 4½ miles of 20-in. main. The losses at the farthest ends of these mains rarely exceed 8 lb. per square inch and the pressure carried was 90 lb. The system was well patronized on account of its convenience for delivering small powers, or in places where the cold exhaust could be used for refrigeration. The trouble from freezing was avoided by passing the air through a coil of pipe heated externally by a charcoal fire. A number of motors of a size exceeding 100 hp. were installed. At this plant it was reported that the cost of compressing 33 lb. of air to a pressure of 90 lb. per square inch was a trifle less than 1 ct.

It is the convenience and safety of the transmission and storage of energy by compressed air which has made it so important and widespread a feature of modern engineering. The convenient return of the exhaust to the atmosphere is in many places an advantage, as in underground or submarine work, and the harmlessness of the air in case of accident, breakage or leakage, is often a valid reason for the use of air engines.

Many of the collieries and mineral mines in this country and abroad have compressed-air transmissions approaching in size the Paris installa-

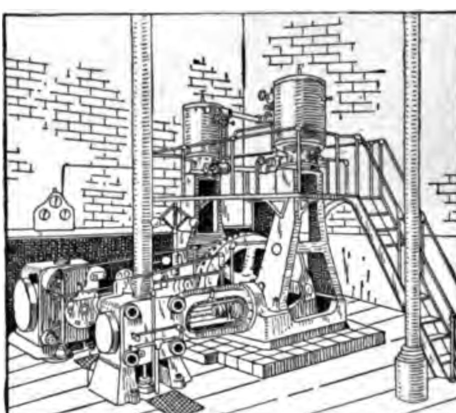


FIG. 270.—Cross-compound horizontal-vertical ammonia compressor.

tion. A coal mine producing 3,000 tons per day will use about 2,000 hp. in compressors and may have from 3 to 10 miles of mains depending on the size and depth of the workings. Some of the copper mines in the upper Michigan peninsula have large compressed-air transmissions, 20-in. pipe being used in a number of cases.

Compressed-air System at Butte and Anaconda.—The Anaconda Copper Mining Co. operates 22 shafts at Butte. In 1912 they commenced using compressed air for hoisting and installed a compressor plant electrically driven. Each compressor has a capacity of 7,500 cu. ft. (465 lb.) of free air (12 lb. abs.) compressed to 90 lb. gage and is run by a 1,200-hp. 2,200-volt synchronous motor. There are eight compressors in all. The system furnishes air for about 40,000 hp. of hoists, the diversity and load factors being low. Seven of these compressors are run continuously the eighth being held in reserve, the excess air being used in the drills. About 13,500 hp. of motors are in use driving smaller compressors in the various mines for furnishing air for drilling and blowing

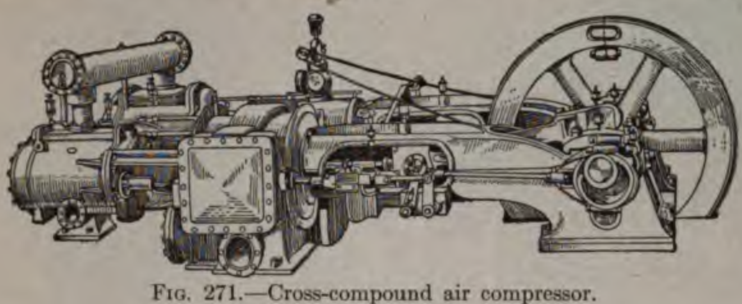


FIG. 271.—Cross-compound air compressor.

out the workings. There is a hydrostatic storage plant for the hoisting service of 66,000 cu. ft. capacity and the air is distributed by something over 3 miles of mains with a pressure drop of about 3 lb. Storage reser-

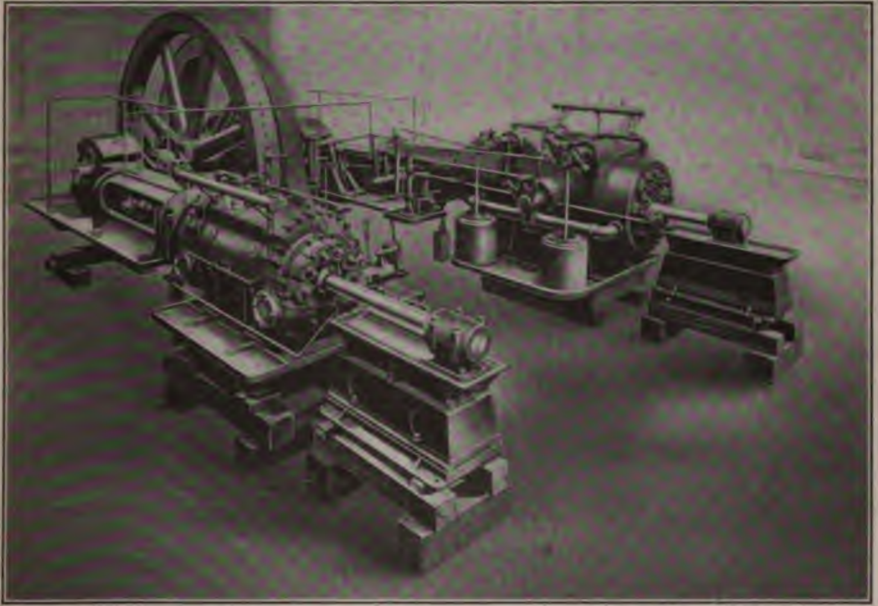


FIG. 272.—Nordberg two-stage motor-driven air compressor, Anaconda Copper Mining Co., 7500 cu. ft. free air per mine.

voirs of about 8,400 cu. ft. capacity are installed at each of the large hoists to prevent excessive loss of pressure in the lines.

After various electric systems had been considered the compressed-air system was installed for the following reasons:

1. Total cost was lower, due to the fact that the existing steam hoists could be readily changed over at small expense for air working.

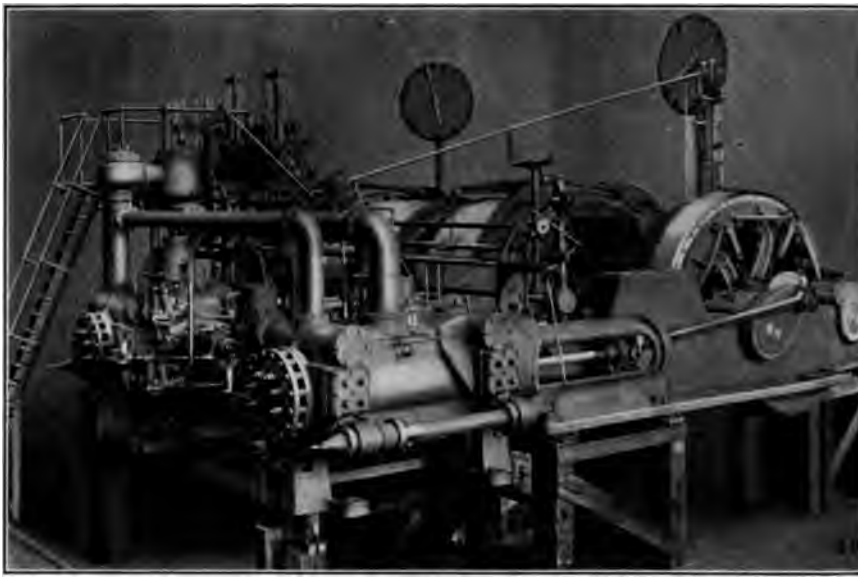


FIG. 273.—Nordberg duplex-geared air hoisting engine, Mond Nickel Co.

2. Large power storage capacity could be cheaply provided to overcome excessive inrush in starting.

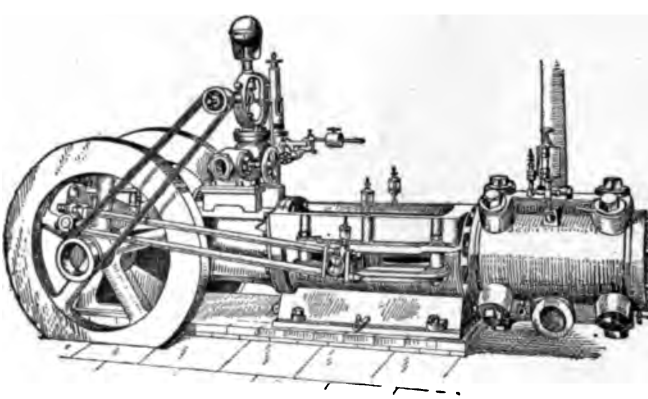


FIG. 274.—Straight line air compressor, Meyer cut off.

3. Synchronous motors could be used in the compressing plant maintaining the power factor and load factor at 100 per cent.

4. Excess air could be used with great advantage in the drill system.

5. A steam drying plant in existence at each mine rendered reheating cheap and easy and largely increased the efficiency.

It was found by test that from 1.4 to 1.5 kw. were used per indicated horsepower in the hoist. See papers by Nordberg, Gillie and Hebgen, *Transactions* A.I.M.E., vol. 46. See also paper by Pauly, *Transactions* A.I.M.E., vol. 42.

The most modern and also the largest installation of this kind is the plant of the Rand Mines Power Supply Co., Ltd., in South Africa, which was installed in 1909–11. The compressor station is located at the Robinson Central Deep, where 12 motor-driven compressors of the centrifugal type are installed, and in addition four turbo-compressors are installed at Rosherville, 5 miles away. Twenty-seven and one-half miles of main

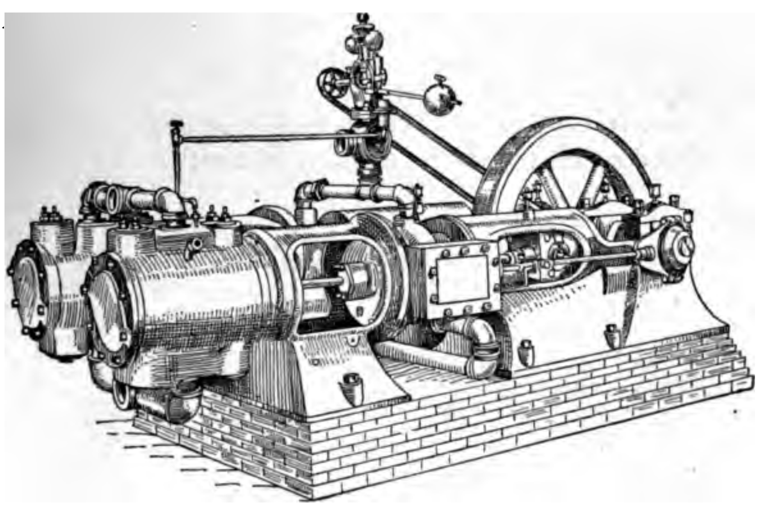


FIG. 275.—Tandem duplex compressor.

piping 27½ to 9 in. in diameter served to distribute the air to the 13 customers whose own mains extend throughout their underground workings. The larger compressors are driven by a 7,000-kw. motor and deliver 2,900 lb. of air per minute at 100 lb. gage. The smaller compressors are half this size. The pipe lines have a capacity of 310,000 cu. ft. and 46,000 lb. of air are thus stored in the lines between the pressures of 90 and 120 lb. (the allowable variation). The yearly output in 1914 was 2,826,500,000 lb. of air at 34 per cent. load factor, with a maximum demand of 15,800 lb. per minute. The air is used for operating small hoists, ore pocket gates, etc., for blowing out the working places after blasting and for operating rock drills, the last being the principle use. Both the fixed- and loose-hammer percussive-type rock drill is used, the rock being too hard for rotary drill. All air is sold on meter readings, the Rand Co. supplying

Venturi meters and the mining corporations swinging gate meters, the mean of the two readings being taken. The air unit is a purely commercial unit and was fixed by local considerations at 27.441 lb. of air at 100 lb. gage pressure, corresponding to 0.641 kw.-hr. See paper by A. E. Hadley, I.E.E., 1913, and paper of J. H. Rider, I.E.E., 1915; also Klingenberg, *Bau. Groz. Elek.*

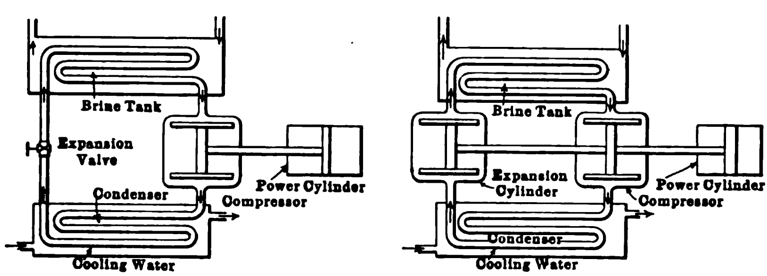
Storage of compressed air in small bulk and with little weight in strong tanks led to its use for street car service as early as 1878, and a number of street car systems were installed. In all cases these systems proved to be cheaper and better than the horsecar system which they displaced. The system usually used the air at about 300 lb. per square inch. A number of reservoirs consisting of Mannesmann bottles about 9 in. in diameter and 6 ft. long located under the seats, held the supply air at a much higher pressure, usually around 2,500 lb. per square inch. Most of the systems included a tank of superheated water, through which the air was passed on its way to the motors. The system was, however, too costly to compete with the electric trolley system and has been almost entirely abandoned in the various localities where it was installed.

CHAPTER XXV

REFRIGERATING MACHINERY

The use of so-called freezing mixtures for the abstraction of heat has been known for many years and is still used for domestic purposes and for a few other applications. Mechanical refrigeration had been in use for about 100 years when the first machine using ether was invented. Since that time air, water vapor, sulphur dioxide, ammonia, carbon dioxide and other fluids have been used as refrigerating mediums, but today only air, carbon dioxide and ammonia are of practical importance. The two chief uses of refrigeration are for cold storage and transportation and the making of artificial ice.

We may classify modern refrigerating machinery as dense-air, compression machines using carbon dioxide or ammonia and ammonia absorption machines. The dense-air machine, used to quite an extent in



AMMONIA OR CARBON DIOXIDE COMPRESSION SYSTEM

DENSE AIR SYSTEM

FIG. 276.

marine practice, consists of a compressing cylinder, in which the air is compressed to about 225 lb., a water cooler which cools the compressed air, an expansion cylinder in which the compressed air is expanded to about 65 lb. and the refrigerating coils, where the expanded and cooled air absorbs the heat. This is known as the dense-air system and apparatus of economical size may be employed due to the high pressures used. This system is largely used for ice making and ice-box cooling on shipboard because of its safety (no dangerous fumes in case of leakage in confined spaces) but it is not efficient.

In the compression systems using CO_2 and NH_3 the gas is compressed to such pressure that when cooled in the condenser by the cooling water it will liquefy. The liquid is then expanded through a valve into the refrigerating coils where it absorbs sufficient heat to evaporate the liquid and the gas is then led to the suction side of the compressor to again begin the cycle.

These machines are much more efficient than the dense-air machine and with CO_2 pressures as high as 900 lb. must be used. They are largely employed in marine practice especially in the frozen-meat trade. An additional reason for their use is the fact that they can be applied to the extinguishing of cargo or bunker fires. The ammonia compression machine is used on land to a much greater extent than the others and more economical results are obtained than with the CO_2 or dense-air systems. The ammonia used is anhydrous and the only serious troubles

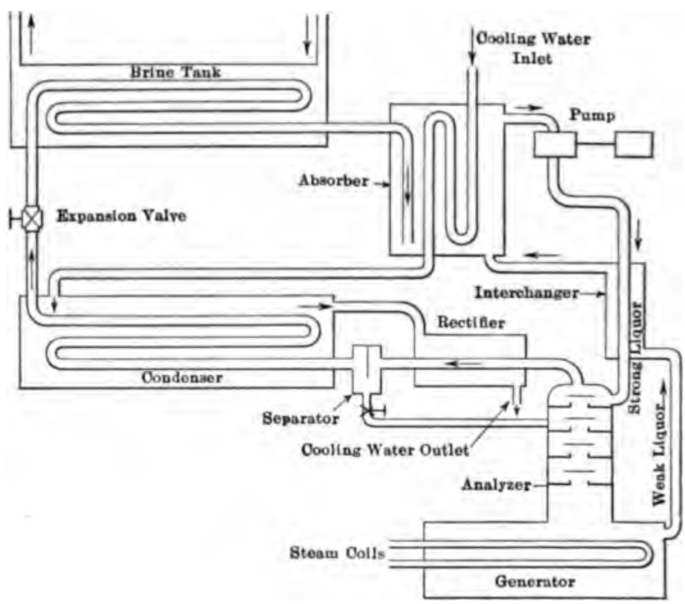


FIG. 277.—Absorption system.

come from leakage of ammonia into the air and water into the ammonia. A tight system will avoid these troubles.

In the ammonia-absorption system a solution of NH_3 in water is used. The strong NH_3 solution is heated by steam coils in the generator, and the NH_3 driven off at a pressure of about 150 lb. The gas passes through the analyzer and rectifier and then to the condenser where the ammonia is liquefied by the cold-water circulation. The liquefied ammonia is expanded through the cooling coils to the absorber in which the evaporated ammonia is absorbed by water thus keeping a low pressure in the coils. The liquor is then pumped to the generator to go through the cycle again. This system is not efficient in small units and is better adapted to refrigerating than to ice making. Many of the large systems in abattoirs, cold-storage plants and central stations for refrigeration are of this type. The efficiency of the ammonia absorption and compression systems are practically equal under commercial conditions.

In the direct-expansion system the refrigerating fluid is circulated in the cold room in pipes but when air is the medium the room becomes part of the system. This system is comparatively little used on account of the regulation troubles, presence of moisture and difficulties of leakage when CO_2 or NH_3 is the medium.

The more usual and better system is the brine circulation system in which the expansion coils are submerged in a brine tank, the cold brine (a solution of salt or calcium chloride) being circulated by a pump through coils in the cold room.

The unit of refrigeration is the "ton of ice melting per 24 hr." The latent heat of ice is approximately 144 B.t.u. so the ton of refrigeration = 288,000 B.t.u. per 24 hr. or 200 B.t.u. per minute. The ice-making capacity is somewhat less than half this figure.

In a refrigerating system the lower temperature is fixed by the room temperature required for refrigeration and the upper temperature is fixed by the amount and temperature of the circulating water. The pressures are fixed when these temperatures and the medium are known. In the American Society of Mechanical Engineers' rating the temperatures are taken as 0° and 90°F. and the economy is taken as the ice-melting effect per pound of coal or per indicated horsepower.

The usual piston displacements in the compressor per ton of rated capacity vary between 3.5 and 5 cu. ft. per minute and the power required varies from 1 to 2.5 i.hp. per ton of refrigeration.

Good average efficiencies are about 25 lb. of ice-melting effect per pound of coal with either compression or absorption system. The dense-air machine is not nearly as efficient say 3 to 8 lb. of ice-melting effect per pound of coal. About twice the theoretical amount of cooling water is required for good work. Practical figures lie between 1 and 3 gal. per minute per ton of capacity.

Ice Making.—Artificial ice, one of the important applications of refrigeration, may be made either by the plate system or can system. In the plate system a series of compartments from 12 to 16 in. wide, 4 to 6 ft. deep and 10 to 20 ft. long, are constructed from cast iron or steel plates behind which the brine circulates. A movable brine circulating coil is sometimes used in the center of the compartment to cool the water to the freezing temperature. After the freezing has begun this coil is swung out of the way. On the completion of the freezing process which may take from 30 hr. to a week warmer brine is pumped through the passages loosening the plate ice and the plate is lifted by a crane and sawed into blocks of suitable size for marketing.

In the can process steel cans holding from 100 to 600 lb. of water are suspended in the cold brine tank. The freezing takes from 2 to 24 hr. and the tanks are emptied and handled by similar machinery to that employed in the plate system. Clear ice is obtained in both systems by a process of agitation.

There is a third system in which a revolving cylinder, in which the brine circulates, dips in the water tank and becomes covered with ice crystals. These are scrapped off and pumped with some water into a hydraulic press which converts the slush into a cake of ice by squeezing out the water. It is difficult to obtain clear ice in this process.

Cold Storage.—The brine-circulation system requires about 50 to 100 per cent. more surface in the pipe coils than the direct-expansion system. For freezing fish and meat the surface may be 100 per cent. larger still. The following table of lineal feet of 1-in. pipe required per cubic foot of cold storage space has been adapted from Siebel.

Size of room in cubic feet		Room temperature, F°. Direct-expansion system					
		0°	10°	20°	30°	40°	50°
100	Average	4.0	2.0	0.5	0.4	0.3	0.2
1,000		1.25	0.3	0.2	0.15	0.1	0.07
10,000		0.8	0.2	0.14	0.1	0.07	0.04
30,000	Insulation	0.6	0.15	0.1	0.07	0.04	0.03
100,000		0.4	0.12	0.07	0.05	0.03	0.02

For brine circulation multiply by 1.75.
For 1¼-in. pipe multiply by 0.8.
For 1½-in. pipe multiply by 0.65.
For 2-in. pipe multiply by 0.55.

Number of cubic feet per ton of refrigerating capacity per 24 hr. Direct expansion. For brine circulation multiply by 0.57.

Size room	0°	10°	20°	30°	40°	50°
100	120	500	650	800	1,300	2,500
1,000	400	2,000	2,500	3,200	5,000	11,000
10,000	600	2,500	3,200	5,000	8,500	16,000
30,000	800	4,000	5,000	7,000	12,000	23,000
100,000	1,200	6,000	8,000	12,000	18,000	35,000

PROBLEMS

87. The poultry, vegetable, meat and wine rooms on a passenger steamer occupy a space approximating 7,000 cu. ft. The dense-air system with brine circulation is used. Assuming that a temperature of 30° is maintained, what will be the amount of 1-in. pipe coil required, the rating of the dense-air machine and the horsepower required?

88. A cold storage company has a building 80 ft. wide, 100 ft. deep and four stories high. Assume one floor for machinery and 10 per cent. of the other floors for elevators and stairs, ceilings 10 ft. high, and a general business requiring an average temperature of 40° in the rooms, brine circulation system. Find the length of pipe coils, rating of the machine, horsepower required and amount of cooling water per day assuming a rise of 30°. Assuming an ammonia-compression system, find the size of compressor if the piston speed is 300 ft. per minute. Estimate the coal used per day.

CHAPTER XXVI

HYDRAULIC POWER

Although the generation of power by heat engines is a development of the last 200 years, hydraulic and air-power have been known and used for a much longer period and their beginnings go back at least to the Christian era. Air-power is of small relative importance, but hydraulic-power, water-power, in favored localities is of great importance and must always be considered by the power engineer.

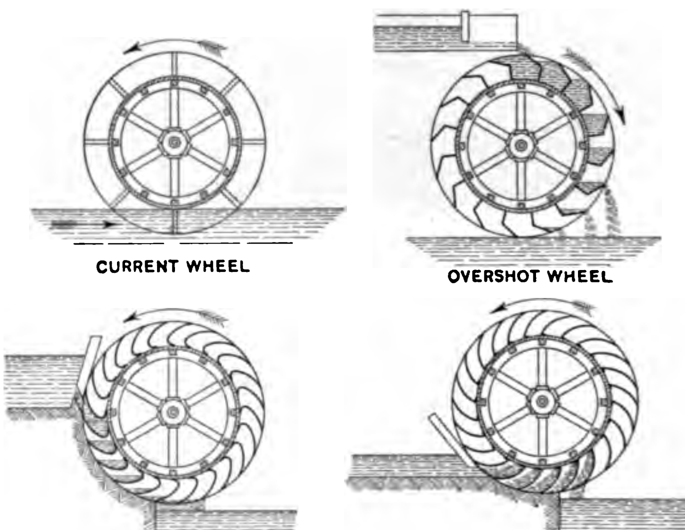


FIG. 278.

The potential energy of water may be measured by its weight (W)—the force available—multiplied by the available fall (h)—the space through which the force is to be exerted—or $E = Wh$. The theoretical power (horsepower) of water in motion is given by the formula hp. = $\frac{WV^2}{550 \times 2g}$ where V is the velocity in feet per second, W is the weight of water per second and g the acceleration due to gravity. For an efficiency of 80 per cent. this formula reduces to $\frac{Qh}{11}$ = hp. where Q = cubic feet per second, h = head in feet, which may be easily remembered for rough calculations.

The earliest waterwheels were "current wheels." These were large wheels, Fig. 278, with paddles which dipped in the stream and were turned by the velocity of the current. They were mainly used for raising water for irrigation purposes or for driving an archimedean screw. The efficiency was very low, 3 to 5 per cent. A few modern current wheels have been built and a little higher efficiencies have been secured.

All current wheels depend on the natural velocity of the stream and an early improvement led the water through an artificial channel or flume where a greater velocity could be secured and the wheel was made with

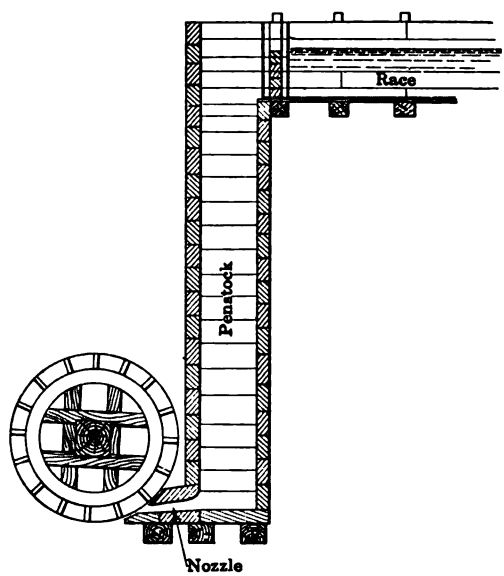


FIG. 279.—Flash wheel.

small clearance at the bottom and sides so that practically the whole of the water was made available to drive the wheel. This improved wheel was known as the "undershot" wheel, Fig. 278, and efficiencies from 25 to 40 per cent. were obtained. Later the bottom of the flume was built up with very small clearances to the height of the center of the wheel making the modification known as the breast wheel, Fig. 278, with efficiencies as high as 50 or 55 per cent. All of these wheels had straight buckets or paddles. Poncelet improved the breast wheel by curving the paddles and making them deeper and by utilizing the breast as a dam and allowing the water to spurt out near the lowest portion of the wheel increasing the efficiency to 60 to 65 per cent.

Many wheels of this type were built in the eastern United States for sawmill work using heads up to 18 to 20 ft. The wheel was usually a built-up wooden wheel with flat buckets about 2 in. deep in a radial direction by sufficient width to give the power required. A rectangular penstock brought the water to the wheel level where the wooden breast and bottom of the penstock formed the nozzle. These wheels were termed "flashwheels," Fig. 279, in the Catskills but were known by other

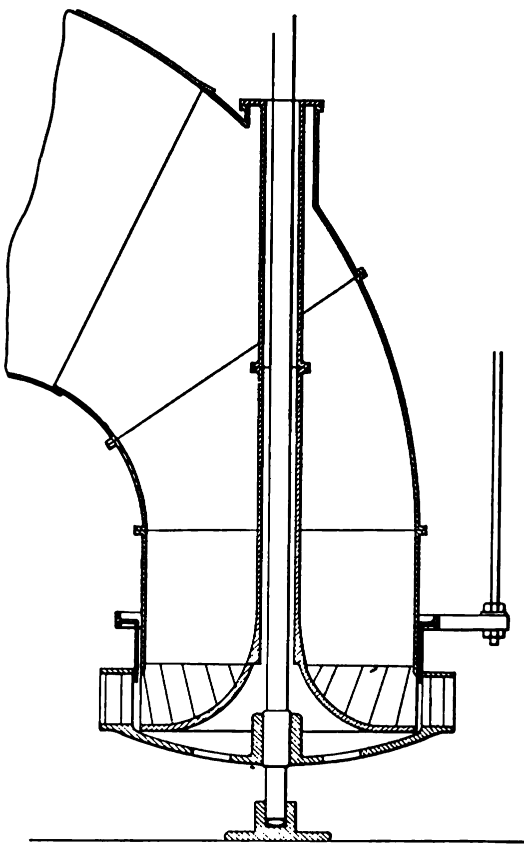


FIG. 280.—Boyden-Fourneyron turbine, Tremont mills.

names in other parts of the country. No good tests of these wheels are extant but fully 50 to 60 per cent. efficiency must have been secured in the better class of wheels.

The "overshot" wheel, Fig. 278, came into use soon after the undershot wheel especially for slower moving mechanisms such as hammers and bellows for the early blast furnaces. The wheel was built with buckets capable of holding the water which were filled from the flume when they were at the top of their travel. The weight of the water turned the wheel and efficiencies exceeding 85 per cent. were often reached. Very large wheels of this type have been built, some of them exceeding 60 ft. in diameter. One of the most famous in America was the "big"

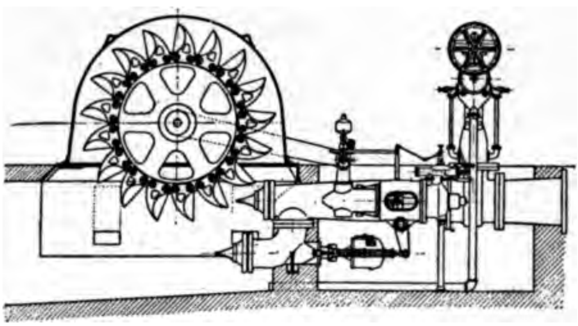


FIG. 281.—14000 H.P. Pelton-Doble water-wheel unit.

wheel of the Burden Iron Works at Troy, N. Y., which was 20 ft. wide, 60 ft. in diameter and produced 278 hp. at 85 per cent. efficiency (see *Proceedings* A.S.C.E., vol. 79, p. 708).

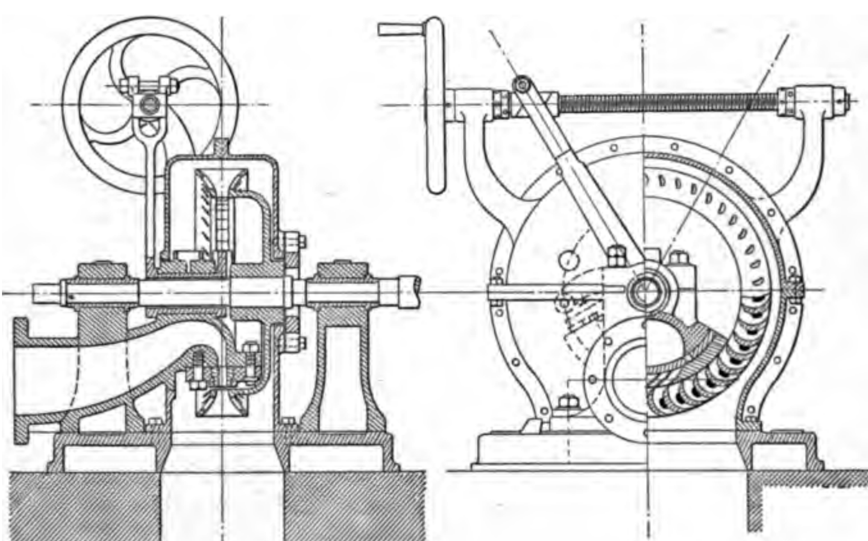


FIG. 282.—"Free deviation" or Girard turbine.

The turbine was invented in France and was introduced into the United States by Elwood Morris of Pennsylvania in 1843, but its development here was largely due to Uriah A. Boyden, who in 1844 designed a 75-hp. wheel for use at Lowell, Mass. (Fig. 280). This was an outward flow wheel but in 1849 J. B. Francis designed for the Boott Cotton Mills at Lowell the inward-flow Francis turbine, now the standard wheel for low-head work both in this country and Europe. These wheels of modern construction give a maximum efficiency of over 90 per cent.

The flashwheel is the probable predecessor of the impulse wheel which came in use shortly after 1870. The impulse wheel is a high-head type and the water is jetted from a nozzle into buckets on the periphery of the wheel (Fig. 281). These wheels are generally known as the Pelton type although the Pelton patent of 1880 was antedated 5 years by the Atkins patent which was not utilized. These wheels give an efficiency of 80 to 85 per cent. A European type of impulse turbine which has been used to a considerable extent is the Girard, also known as a free deviation turbine (Fig. 282). The water is led inside the ring of buckets and jetted outward. High efficiencies have been obtained.

Water turbines may be classified as impulse or reaction turbines with more justice than steam turbines, but the terms, partial intake or full intake, better describe the classification. They may also be classified as to the direction of water flow as axial or radial flow, and inward or outward flow, but practically all modern turbines fall into two classes, the inward-flow central discharge turbine, known generally as the Francis turbine, and the class in which a free jet impinges on an open bucket generally known as the tangential or Pelton type.

With the current wheel no serious constructions were required as the wheel was supported by floats or cribs in the moving current of the stream, but with the better types of machinery flumes and masonry supports had to be constructed, dams became necessary to artificially increase the available head and to store water to secure a uniform flow. The study of the variation of stream flow became a necessity and rainfall and run-off records were kept and compared. These records are now made and published by the Government in the Water Supply Reports for the run-off of streams and by the Weather Bureau for rainfall.

To secure stream flow or run-off records the stream must be rated and a gage maintained and read at least once each day usually at 8.00 a.m. A cross-section of the stream is chosen where the river is straight for a considerable distance on both sides and where the bottom and sides are rock or hard gravel or hard clay not liable to change during floods or low water. This cross-section is accurately surveyed and a gage is established with its zero at the lowest low-water level on record.

Next the velocity of flow of the stream is obtained at as many gage readings as possible. This is usually done by means of a current meter (Fig. 283). The cross-section is divided into many small areas and a velocity reading is obtained at the center of gravity of each area. These readings are averaged for the mean velocity and the discharge is calculated in cubic feet per second for that gage reading. Where a current meter is not available floats or rods weighted to float in an upright position may be used; the velocities being obtained by timing the floats in passing a given distance downstream. After the discharges have been obtained for a number of gage heights a rating curve may be plotted connecting each gage height of the stream with a discharge. When the daily discharges are plotted a curve known as a hydrograph is obtained in which all the variation of the flow of the stream is shown graphically. The daily discharge is always given in cubic feet per second or cusecs. The aver-

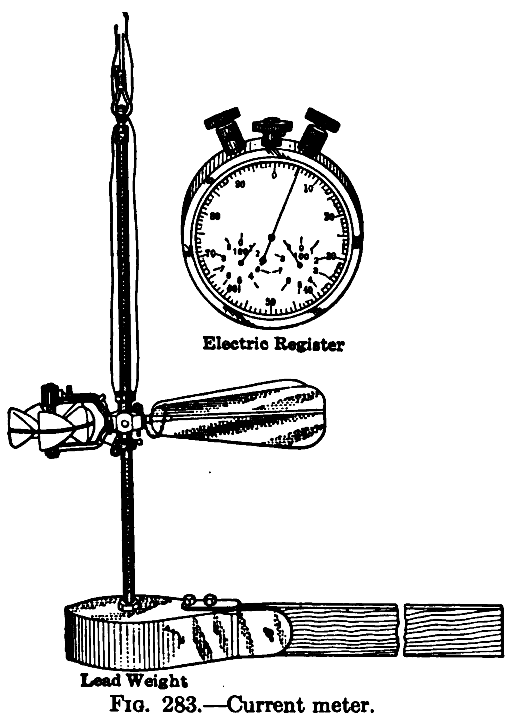


Fig. 283.—Current meter.

age yearly and monthly discharges are also given as inches of water on the watershed or catchment area in order that they may be compared with the rainfall. Maximum and minimum discharges are also noted and stated in cubic feet per second and also in second-feet per square mile of catchment area.

There have been a great many attempts to connect rainfall with run-off so that in the absence of long term gaging records nearly correct figures for a given watershed might be obtained from the rainfall records which usually cover a much longer period. There are a number of papers and much discussion of this subject in the *Transactions* of the A.S.C.E., 1913–14 and 1915, but this method should be used with discretion and only when gagings are not available.

It should be remembered that while the water flowing in the streams is due to rainfall, some portion of this is evaporated, much is absorbed by vegetation at certain periods of the year, and a considerable fraction may be stored in the earth for long periods. Many experiments have been made to ascertain these quantities under certain conditions and details may be found in Rafter, "Hydrology of New York;" Mead, "Hydrology;" and the *Transactions* of the A.S.C.E.

Having the gagings of a stream for a number of years the mean monthly flows can be calculated and a curve plotted showing the summation of these flows. This curve is called a mass curve and examples are shown in Figs. 284 and 285. A straight line passing through the origin and tangent

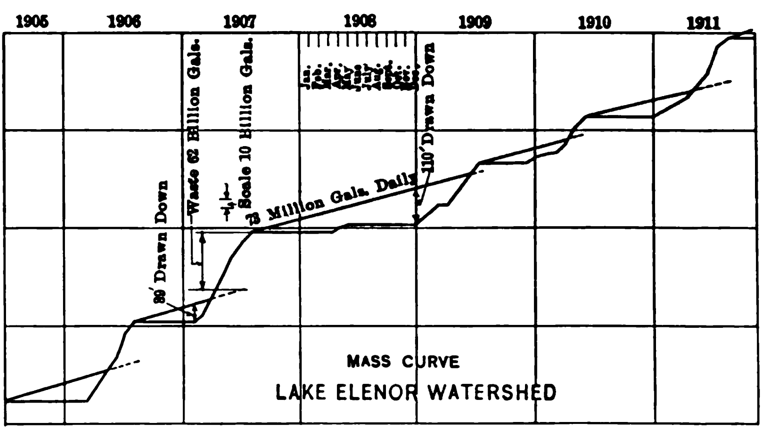


FIG. 284.

to the lowest portion of the curve will give the largest average flow to be obtained from the stream and the necessary storage can easily be calculated. It usually will not pay to provide this amount of storage but a discussion of the mass curve and a survey of the available locations for storage reservoirs and dams will soon show the economical size of storage and the economical mean flow to be provided for. The longer the set of gagings the better will be the work especially if they include a minimum year. It is interesting to plot the variations of the yearly rainfall for the available years as a curve in conjunction with the hydrographs. A study of these curves for any watershed will well repay the trouble. It is, however, important that a sufficient number of well-located rainfall stations be present on the watershed or the curves will be misleading.

The creation of a large storage reservoir is usually a costly operation both as to dam and land damages. The following table of the cost of

large storage reservoirs has been compiled from various authorities and shows the extreme variations of cost. Compare the differences of the cost of the Ashokan and Croton reservoirs on small streams with the cost of the Assuan storage on the Nile.

COST OF STORAGE RESERVOIRS

Location	Storage, billions of cubic feet	Cost per billion, cubic feet
Assuan, Egypt (new)	81.0	$238,000
Assuan, Egypt (old)	35.0	343,000
Ashokan, N. Y.	16.0	792,000
Christiana & Harts River, Transvaal	11.45	1,560,000
Belle Fourche, S. D.	9.36	94,000
Wachusett, Mass.	8.42	269,000
Aziscohas, Maine	8.0	125,000
New Croton, N. Y.	7.84	973,000
Chattanooga, Tenn. (approx.)	7.5	533,000
Buena Vista, Cal.	7.4	21,000
Laramie River, Wy.	5.23	23,000
Indian River, N. Y.	4.46	19,000
Croton, N. Y.	4.27	972,000
Lake MacMillan, Pecos, N. M.	3.88	47,000
Bear Valley, Cal.	1.74	39,000

Dams may be classed as:

1. Earth.
2. Earth with core wall.
3. Hydraulic-filled earth.
4. Timber.
5. Masonry.
6. Concrete and cyclopean masonry.
7. Hollow, Ambursen or reinforced concrete.
8. Steel with or without rock fill.
9. Moveable or Bear trap (barages).

The earth dam is a simple bank of earth. The top soil is usually cleared away to good firm soil and then the dam is built in thin horizontal layers well watered and compacted by rollers as the work progresses. The upstream slope is usually 4 or 5 to 1 and protected by riprap or pitched with cobble or flagstones. The downstream slope is 2 or 3 to 1 and is sodded. These dams should not be built over 8 to 10 ft. in height if they are expected to be tight and the width of the horizontal top should be equal to the height. Ample spillways of masonry or solid timber should be provided. In some localities such as irrigation work in India these dams have been built much higher and the percolation through the dam has been regulated with good success.

Earth dams for higher heads should be constructed with core walls to prevent seepage and percolation which would eventually lead to the destruction of the dam. The core wall may be of puddled clay, timber sheet piling, stone, concrete, or steel. The core should start a sufficient distance below the foundation of the dam to prevent dangerous seepage. The core must be thick enough to be impervious and should be carried up nearly to the crest of the dam and above the spillway level. Many experiments have been made to determine the line of ground water in an earth dam but this seems to depend on the nature of the materials, which should be thoroughly investigated before construction and all improper earth thrown away.

At Gatun, Panama, the 30-mile lake of the Panama Canal is held back by the Gatun dam, a hydraulic-filled dam with rock toes and riprap facings. The toes were first placed of heavy rock and then soft mud and sand pumped in to fill the interior as the riprap slopes were carried up. This dam is 120 ft. high and about a mile wide. It sustains a head of water exceeding 80 ft. The Necaxa dam of the Mexican Light and Power Co. is of this type and is 190 ft. high, the highest earth dam in the world. It has a puddled clay core, riprap slopes and a hydraulic fill.

Timber dams are built by sinking square cribs of timber which are filled with riprap to hold them in place. On these cribs the covering planks are laid. Another good way on a small stream is to drive a line of wooden sheet piling across the stream. To the sheet piling is spiked a 12- by 12-in. mudsill as low down as possible on the upstream side. Cribs of round poles are placed downstream of the sheet piling supporting the wales and the cover planks are spiked to both mudsill and wales. The dam soon silts up on the upstream side and remains tight as long as the silt is there. The whole length of the dam is usually the spillway in which case an apron of planking is necessary on the downstream side to take the impact of the spill water and prevent undercutting.

Masonry dams built of brick or cut stone are of all sections, the plain rectangular wall with a capstone sloping downward upstream being very common for low heads. The best form for a masonry dam is dependent on the kind and weight of the masonry and whether it is also to be used for a spillway. For this purpose the ogee form is the most frequently adopted as giving the maximum effect with the smallest cost and such dams are often submerged 10 to 15 ft. without danger. Masonry dams are rarely built except upon a rock foundation.

Concrete may be used instead of masonry or the so-called cyclopean concrete in which very large stones often exceeding 10 tons are imbedded in the concrete. It is said to be important that these stones do not touch but at least one large dam was constructed by piling up the large stones in the forms and then grouting the pile thoroughly. This construction cannot be recommended. Small dams have been built of reinforced concrete in sections similar to a retaining wall but for large constructions and a solid dam steel reinforcing is not necessary. All masonry and concrete dams should go deep enough into the solid rock to prevent failure by shear or moving bodily downstream. In the dam at Austin, Texas, which failed by moving downstream, the toe was at the upstream end of the section and was only 24 in. wide. The dam at Austin, Pa., failed in the same manner.

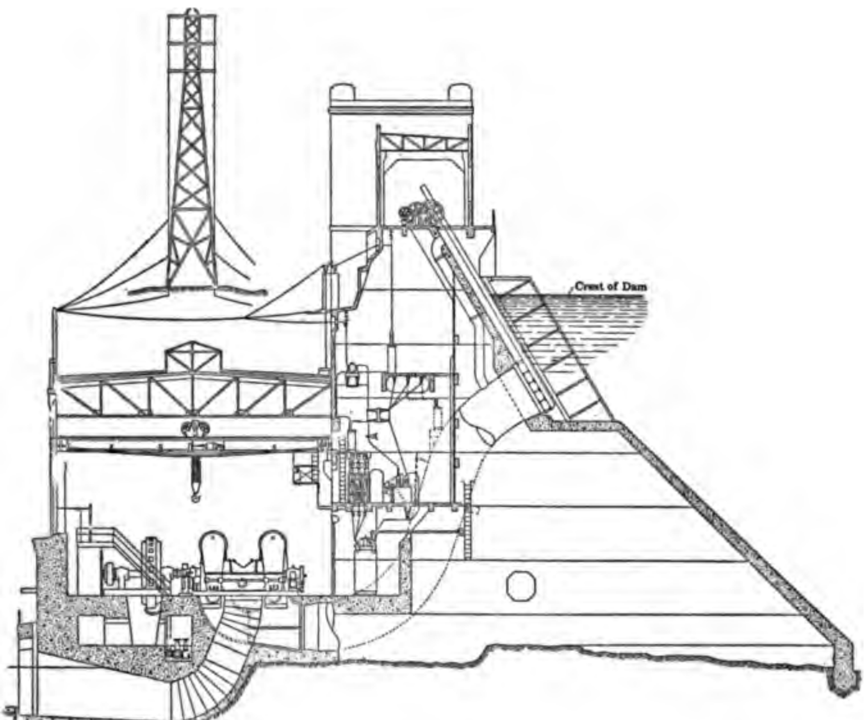


FIG. 286.—Section of Estocada dam and power house, Portland Railway Light & Power Co., Ore.

The hollow or Ambursen concrete dam is usually built "A" section in bays of proper width and the usual design of reinforced-concrete structures is followed. The design has been criticised by some engineers who maintain that concrete under water is not sufficiently impervious to preserve steel from deterioration but the oldest Ambursen structures have shown no sign of deterioration up to date. In this type of dam the power house may be placed in the interior of the dam and remarkably economical construction results.

Steel dams are of two types. The commoner consists of a set of "A" frames with buckle plates riveted to the upstream flanges. These plates may be protected from deterioration by a thin layer of concrete. The second type has been used in the rocky canyons of the Western States where a steel-plate core has been anchored in the side walls and bottom of the canyon and broken rock has been dumped on both sides of the plate to provide stability. In some cases the plate has been protected by a light concrete wall on both sides.

The bear-trap or movable dam was developed during the canalization of the European rivers and has been extensively used by the United States Government engineers on the Ohio and its tributaries in the improvement of navigation. During a flood the dam is lowered to the bottom of the river but as soon as the flood subsides the dam is raised and a pool sufficient for navigation is formed. The bear-trap dam consists of a number of units or palets about 24 in. wide and the height of the dam. These palets are hinged at their bottom ends to a heavy masonry construction at the bottom of the river. At the point of center of pressure at the downstream side a strut is attached, which fits in a lock at the bottom of the river below the hinge and holds the palet in a nearly vertical position. In another type the strut is hinged at the downstream end and the connection of strut and palet is so placed that the unit when set up is stable until the water reaches a certain height when the palet is overbalanced and falls to the bottom. Other types have "A" frames supporting a platform and longitudinal girders on which square logs of wood are supported. These are removed by the attendants in time of flood and replaced afterward.

Spillways.—Every dam should be provided with a spillway of sufficient capacity to take care of the maximum flood. Frequently the whole crest of the dam is used as a spillway as at Holyoke and Hales Bar and such dams must be made heavy to prevent overturning when the water is at its maximum height above the crest. Where the floods are of smaller moment a portion of the crest is made lower and walled in at the sides to act as a spillway as at the Delta and Hinckly dams, N. Y. The spillway is often placed at a distance from the main dam where rock is available and a first-class construction secured at less cost, as at Ashokan, N. Y. Where floods are severe and of very rapid rise it is common to provide a small spillway for normal use and to install gates of sufficient size in the dam itself to take care of the flood waters. Examples of this type of construction may be seen at Chevres on the Rhone, Switzerland, where large gates of the "Stoney" type are in use and at the Scotland Dam on the Shetucket, Conn., where gates of the "Taintor" variety are installed. It is usual to allow a rise of about 5 ft. over the crest of the spillway at maximum floods.

During ordinary seasons the crest of the spillway is increased in height by flashboards. The simplest construction consists of pieces of ordinary pipe 10 ft. apart imbedded about 1 ft. in the concrete crest of the dam. Into these pipes, pieces of round iron 1 in. to 1½ in. in diameter and extending in some cases 3 ft. above the crest, are inserted. On the upstream side of these rods pieces of 2 by 12-in. planking are laid on edge thus raising the crest 3 ft. (see Fig. 287). The planks are held in place by the water. In case of a sudden flood topping the crest the iron rods bend over, allowing the planks and water to escape down the spillway. There are a number of patented flashboard constructions but the commoner type is the best and cheapest.

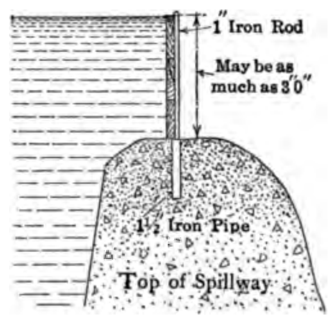


Fig. 287.—Flashboards.

Hydraulic-station Layouts.—Hydraulic-station layouts are almost always determined by the physical conditions of the case in question, since the local geological conditions usually determine the site of the dam and power house, the quantity of water and head available. At times there may be a choice of types, but this usually happens only when the head is on the dividing line between the Pelton and Francis types, or when the head is so low that a number of wheels must be used on one

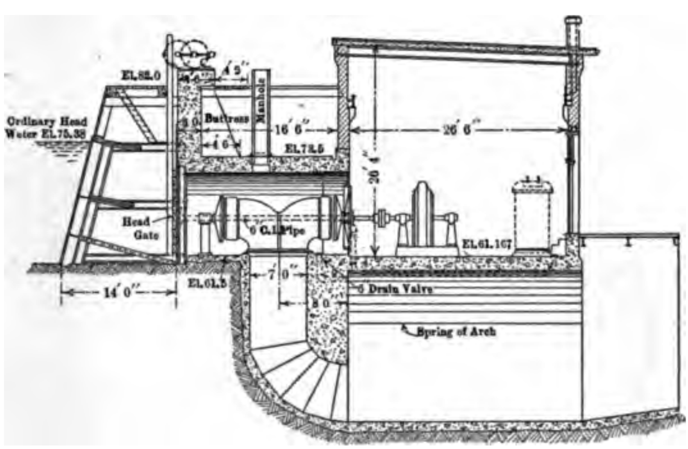


Fig. 288.—Section through penstock and power house, Uncas Power Co., Scotland, Conn.

shaft. There always remains, however, the choice between horizontal and vertical shaft wheels.

Where the stream regimen is fairly constant and the power house is a part of the dam itself three arrangements should receive careful consideration; the horizontal shaft wheel extending into the head race with the shaft parallel to the flow of the stream, and the draft tube to carry the

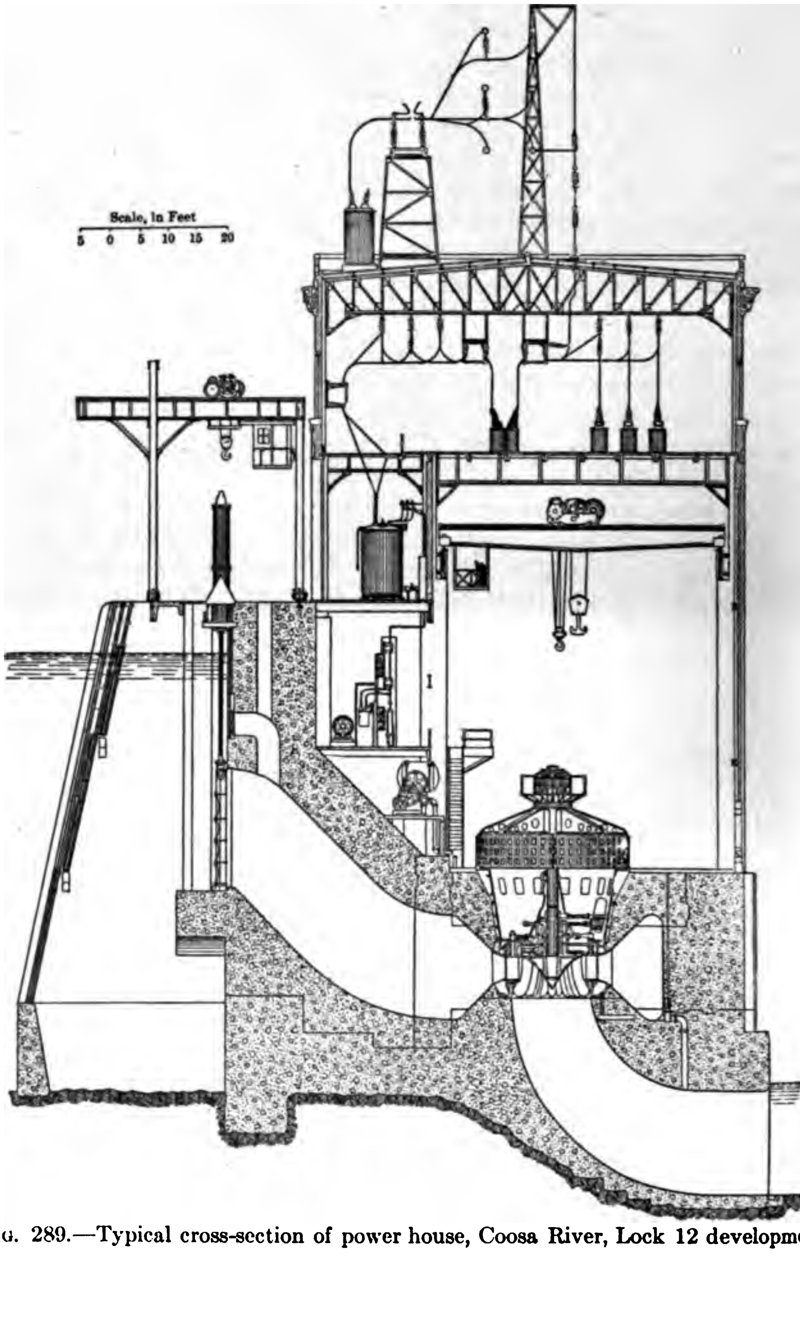


FIG. 289.—Typical cross-section of power house, Coosa River, Lock 12 development.

water away; the horizontal shaft wheel with the shaft at right angles to the flow of the stream, necessitating a penstock and with the draft tube as before. Both of these plans necessitate a power house below the dam, although the Ambursen type, with the power house inside the dam, may be used. Third, the vertical shaft wheel, either submerged or with a draft tube. Here the power house may be above the crest of the dam if desirable. In the first and third cases the penstock is an open one. In the second case the penstock is very short, sometimes only a foot or two longer than the wheel casing. Due to the short connection, there is no trouble from surges or water inertia. With higher heads, or where the power house is located at some distance from the dam, penstocks with or without canals or flumes, are necessary, and either the vertical or hori-

Fig. 290.—Stoney gates, Manchester ship canal.

zontal shaft wheel can be used. The two or three permissible arrangements are usually laid out and quick preliminary estimates are made to determine the best plan. Where plenty of water is available with low heads two or even three or four wheels may be used on one shaft, or a number of wheels may be geared to one shaft. Where there are two good ways of solving the problem it is sometimes advisable to secure proposals on both types of installation and use the most advantageous.

Due regard should be given to the convenience of repair, which may well be in certain cases the deciding factor. There are certain plants in which it is necessary to draw down the pond 10 ft. or more in order to make repairs on one wheel. That plan is usually the best which puts the wheel above the tail water and provides a gate between the head race and the wheel. In good installations a turbine can be opened, examined and put back into service in less than an hour. This is not usually possible with submerged wheels. Large headgates cannot be handled quickly and pumping a wheel pit is a slow and troublesome operation.

Many installations must be built with wheel pits and submerged wheels. In such cases the headgates should be very carefully designed and, if large and heavy, cranes should be provided for their rapid and efficient handling. Up to a width of about 5 ft. and a head of 8 to 10 ft., the ordinary wooden stop logs make the best and cheapest headgate for concrete or stone constructions. Above these dimensions steel gates of the Stoney type with roller supports may be used (see Broome gate, Fig. 291), but the construction should be solid and in larger sizes should

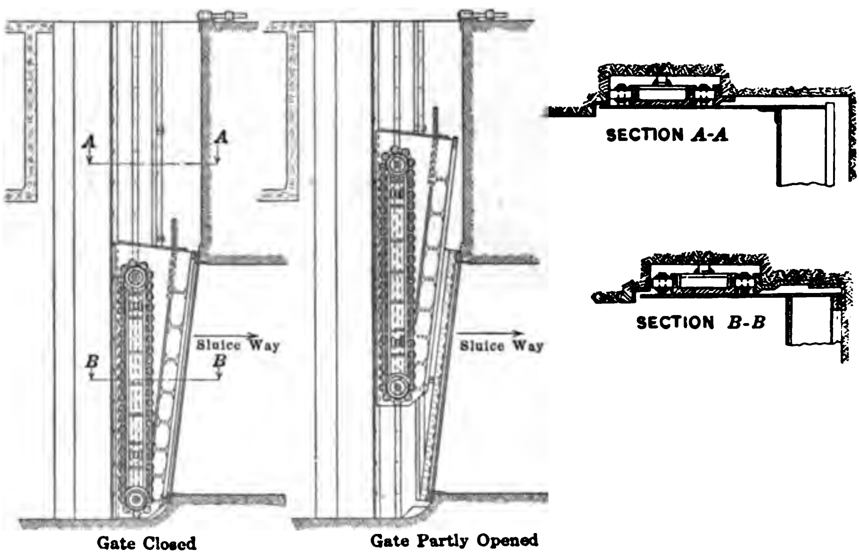


Fig. 291.—Broome gate.

be of the cellular construction used in lock gates. Where rollers are used, cast-iron roller races should be bolted to the concrete or stonework. The smaller sizes of gates are worked by the ordinary screw and winch handles. Many large installations are provided with the Taintor gates for use as headgates. These gates consist of a 30° to 60° section of the surface of a cylinder and are hung on a horizontal axis on the downstream side. They are sometimes as wide as 20 ft. and may be 8 or 10 ft. high, with leather or rubber washers on the sides and bottom of the gate to keep them watertight. They are handled with the ordinary screw control in the smaller sizes, or by a hoist for the larger sizes. At the admirable water-power installation at Chevres on the Rhone below Geneva these gates are used to close the wheel pits on the upstream side, and the sump pump is of such a size that it is possible to examine the three wheels on one shaft and get them back into service in less than 2 hr.

Most rivers and lakes contain more or less floating trash, sticks, leaves, logs of wood, and, in the winter, ice. Suitable means must be provided to prevent this rubbish from entering the penstocks and wheels. Floating débris is usually controlled by a floating boom which directs this class of rubbish over the spillway. The rubbish which passes below the boom must be caught on a rack or screen. These racks are usually made of flat bars, set edgewise and properly spaced and stayed. They must be kept clean, which is usually done by means of a rake and cheap labor. Occasionally, as at Chevres, where the trash is troublesome and the rack is over 300 ft. long, a power rake electrically operated is used. This rake

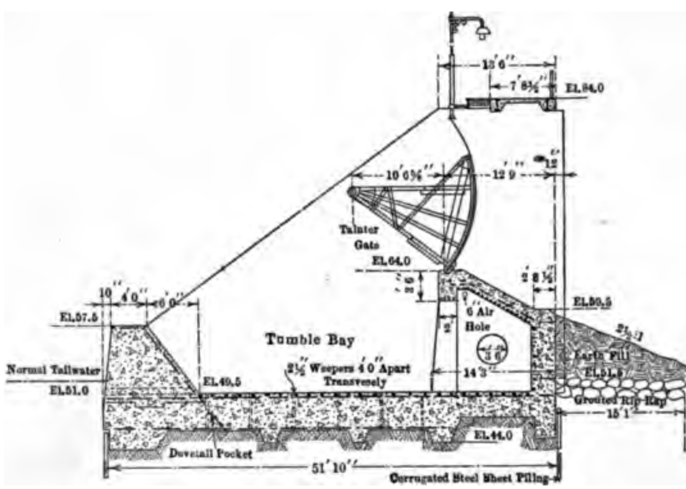


FIG. 292.—Tainter gates, Scotland dam.

is on a car which travels the entire length of the screen, raking and delivering into dump cars. The trash in this case has a fuel value and is dried and burned. In northern localities the same booms and screens are used to divert ice, and in some situations the raking must be continuous during the winter. At Rochester, on Brown's race, the raking for three 600-hp. wheels often requires three shifts of eight men for days at a time. In this race the ice forms on the bottom of the race as well as at the water surface, which makes the trouble very serious. "Anchor ice" is ice that forms on the bottom of passages or races. "Frazil ice" is fine needles of ice, which in certain localities form all through the body of the stream, and when caught in the wheel passages may freeze solid.

Reserve.—When laying out a hydraulic power plant the question of reserve must be considered in connection with the expected load, the water available, and the diversity of water-supply peak and load peaks. As a general rule, one unit of the largest size should be installed as reserve. Where no water is available for reserve or where the minimum water does not coincide with the minimum peak, a heat-engine reserve or standby plant may be necessary. Many hydraulic installations require this heat-engine reserve, and where the variation in water supply is considerable, the steam reserve may be nearly as large as the water power itself. At Utica the Trenton falls plant of 6,000 kw. has a steam auxiliary of 4,000 kw. and both plants will probably be increased in the future in about the same proportions. There are many cities whose lighting and traction supplies are generated by water power, which have steam auxiliaries capable of carrying the full load, in case of breakdown or low water. The Rochester company, taking 10,000 kw. from Niagara Falls, keeps the same capacity in steam power with banked fires as a standby.

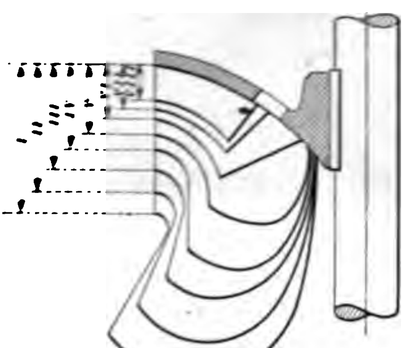


FIG. 293.—Gelpke's standard types. Kaplan's types are similar.

Frequently water power has been sold for delivery only when water power is available. This is known as secondary power and of course brings a lower price. There are cases where tertiary power is also sold, which may be cut off by the hydraulic company at any time it sees fit.

Water Storage Batteries.—It has often been proposed to pump the tail water back into the pond by cheap machinery, or in the time of low load, and to utilize it again at the peak. At the Rheinfall at Newhausen such a water storage is in use. Here a fall of about 60 ft. is available but owing to state restrictions only a certain portion of the water may be used. The load curve is of the usual light power and traction type, with about 50 per cent. load factor. Two combination units have been installed, each consisting of a high-head centrifugal pump, a water turbine and an electrical unit capable of being used either as a motor or generator, all on the same shaft. The turbines in the main plant, at time of light load, furnish electrical energy which is used by the motor and centrifugal pump portion of the unit to pump water to a storage reservoir situated in the hills about 240 ft. above the station. When the peak comes on, the cycle is reversed, and the water in the reservoir is used in the turbine, driving the electric machine as a generator to supply the additional power. There are a number of these installations in Europe, and the idea seems to have originated with Sulzer Bros. who have furnished the pumps and have charge of the installations; Escher, Wyss and Co. furnishing the turbine and Brown, Bouverie and Co. the electric apparatus. It has also been proposed to use the Humphrey pump to pump the tail water back into the pond, but no installations of this kind have been made.

Design and Proportions of Turbines and Runners.—Gelpke in "Turbinien und Turbinienlangen" has given particulars of the design of eight standard types of Francis wheels which are tabulated below with changes to suit American practice (see Fig. 293).

Type	C	Rev. at 1-ft. head × diam., mD.	$\frac{V}{D^2}$	$\frac{P}{D^2}$ $\eta = 0.8$	Gelpke's max. value of η	Moody's best values of η (see curve)	$\frac{B}{D}$	Diam. exit edge of vanes, D
VIII	10.8	77	0.222	0.020	0.83	0.885	0.08	0.66
VII	13.1	80	0.302	0.027	0.84	0.89	0.10	0.67
VI	16.6	83	0.441	0.040	0.845	0.903	0.125	0.70
V	22.2	89	0.685	0.062	0.86	0.915	0.16	0.75
IV	29.4	96	1.03	0.094	0.87	0.918	0.20	0.86
III	40.6	107	1.58	0.144	0.87	0.918	0.25	0.97
II	54.5	121	2.23	0.203	0.83	0.91	0.30	1.10
I	70.4	138	2.87	0.260	0.77	0.87	0.35	1.23

As the speed n, horsepower, hp. and head H are known, C or the "unit speed," *i.e.*, the speed which under 1-ft. head will develop one hp.,

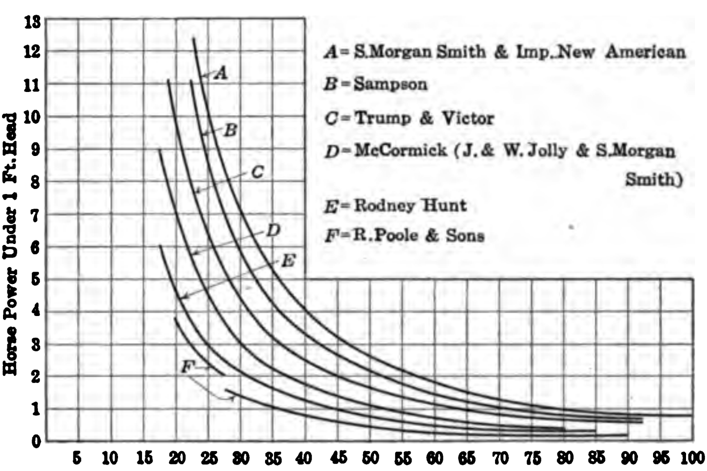


FIG. 294.—Characteristic curves of standard American turbines from catalog figures

$C = \frac{n\sqrt{\text{hp.}}}{H^{5/4}}$ can readily be found and the type determined. If Q is the cubic feet per second passing through the wheel when developing thus: . . . . . horsepower at H head, the quantity passing at 1-ft. head is q and $\frac{q}{11} = P$, the horsepower at 1-ft. head, if the efficiency $y = 0.80$.

Let us suppose a turbine is required to give 500-hp. at 300 r.p.m. under a head of 50 ft. C will then be $\frac{300 \times \sqrt{500}}{50^{5/4}} = 50.4$. This will indicate that a turbine of type intermediate between II and III is required. Taking type II, mD is 121 and $\frac{121 \times \sqrt{50}}{300} = 2.85$ ft. $= D$.

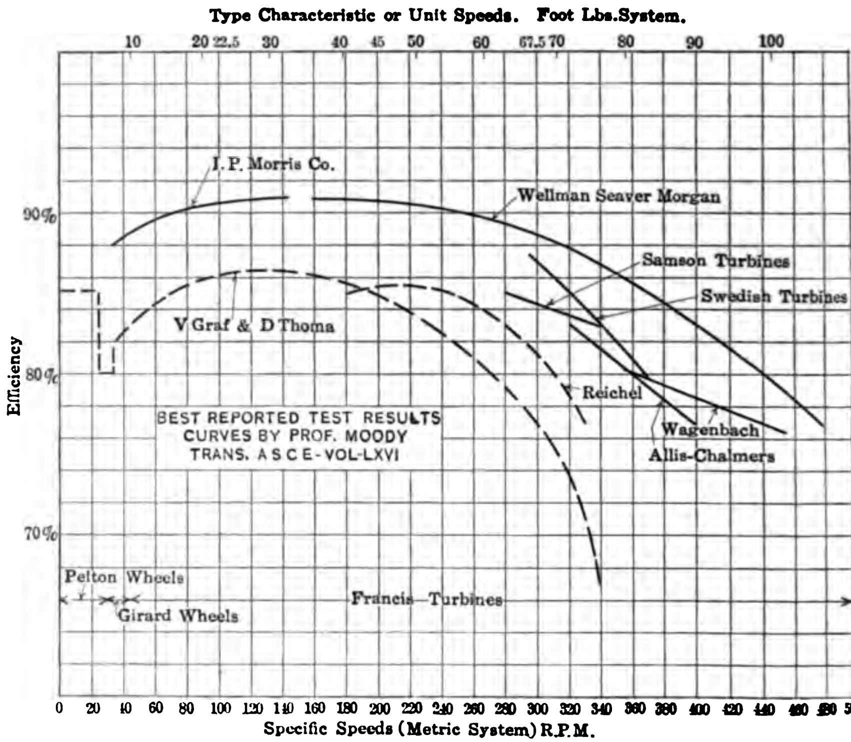


FIG. 295.

A wheel of this diameter at 300 r.p.m. and 50-ft. head will develop about 580 hp. at 80 per cent. efficiency. $\frac{P}{D^2} = 0.203$; multiplying this by D^2 and $H^{5/4}$ gives 583 hp.

Taking type III, mD is 107 and $\frac{107 \times \sqrt{50}}{300} = 2.52$ ft. diam. $\frac{P}{D^2}$ is 0.44. $144 \times 2.52^2 \times 50^{3/2} = 324$ hp. This wheel is much too small. It will probably be possible to find a stock wheel which will more closely fit the conditions than type II.

D. W. Mead has plotted the characteristic curves of the various makes of stock American wheels in his discussion of Larner's paper in vol. 66 of the *Transactions* of A.S.C.E. These curves are plotted with r.p.m. per minute under 1-ft. head as abscissæ and hp. under 1-ft. head as

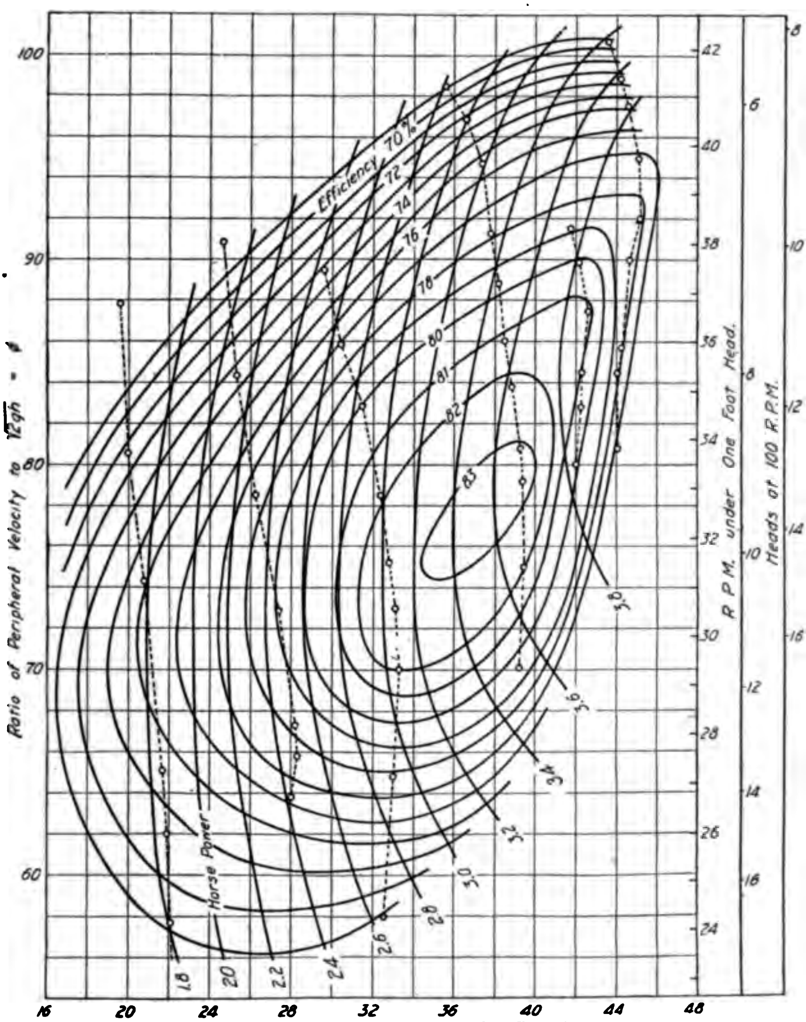


FIG. 296.—Characteristic curve of a 44-in. improved New American turbine. From Mead Water Power Eng.

ordinates. A modified reproduction of these curves is given in Fig. 296. If the efficiency is 80 per cent. the ordinates of these curves multiplied by 11 will give q, the quantity of water used at 1-ft. head and q at any other efficiency may be readily found.

If the "unit speed" $C = \dfrac{n\sqrt{\text{hp.}}}{H^{5/4}}$ and the head H are known it will be possible to pick out innumerable combinations of power and speed depending on the size of the runner used.

Values of C from 1 to 6 or 7 indicate the Pelton or tangential type; from 10 to 100 the Francis type. The values between 7 and 10 indicate the field of the outward radial-flow turbine of the Girard type often called "a free deviation turbine" and hardly known in America.

The coefficient of peripheral speed $\phi = \frac{\text{peripheral speed of runner}}{\sqrt{2gH}} = \frac{0.00653Dn}{\sqrt{H}}$ is a very convenient runner characteristic indicating whether certain high or low values of C are due more to a high or low speed (large or small bucket angles) or to a high or low capacity. According to Zowski for low-speed runners $C = 10$ to 28, $\phi = 0.58$ to 0.7.

Thus from $C = 28$ to 60 would indicate medium-speed wheels with values of ϕ from 0.6 to 0.8. High-speed runners would have $C > 60$ with values of $\phi > 0.7$.

The square of C is called by Mead K_5 = specific speed and by Parker, the type constant. In the *Journal of Electric Power and Gas*, May 25, Aug. 3 and Nov. 9, 1912, G. J. Henry, Jr., presents some interesting tables on both tangential (Pelton) and Francis wheels. He divides the Francis wheels into five types as in the following table:

Types	A	B	C	D	E
Unit speed C	80–68	68–50	50–38	38–24	24–10
Peripheral speed	82–79	79–73	73–68	68–64	64–60
R.p.m. at 1-ft. head	32–155	26–160	20–160	15–160	15–160
q	2–50+	1.25–45	0.8–40	0.5–38	0.3–22
Efficiency: full gate	0.72	0.77	0.80	0.81	0.80
Efficiency: ¾ gate	0.75	0.79	0.82	0.83	0.82
Efficiency: ½ gate	0.68	0.73	0.76	0.77	0.76

For the Pelton wheels he gives the following table where D = pitch diameter of buckets, d = jet diameter.

$\frac{D}{d}$	10	11	12	13	14	15	16	18	22	26	30
C	5.35	4.85	4.46	4.11	3.83	3.56	3.35	2.97	2.43	2.06	1.79
Efficiencies											
Full gate	0.60	0.70	0.76	0.79	0.80	0.81	0.82	0.82	0.82	0.81	0.81
¾ gate	0.80	0.83	0.81	0.85	0.86	0.87	0.86	0.86	0.85	0.84	0.83
½ gate	0.68	0.75	0.80	0.82	0.84	0.84	0.84	0.84	0.82	0.80	0.77
¼ gate	0.62	0.72	0.77	0.80	0.82	0.82	0.82	0.81	0.80	0.77	0.73

ϕ for tangential wheels varies between 0.40 and 0.47. Parker in his "Control of Water" gives good methods of checking up designs of both types of wheels.

Moody in his discussion of Larner's paper gives some curves of maximum efficiencies attained by modern waterwheels of the Francis and tangential types. These curves slightly modified are reproduced in Fig. 295.

The detailed design of a turbine and casing is mainly concerned with the angles of entrance and discharge of the vanes or buckets in the runner and of the direction of the entering water from the guide vanes. This is a highly specialized subject and beyond the scope of this book. Zowski in *Engineering News*, Jan. 6 and Feb. 10, 1910, gives perhaps the best treatment of this subject in English although Parker in his "Control of Water" gives the treatment in good form. The general turbine equation may be written:

$$2gH = (U^2 - U_2^2) + (W^2 - W_2^2) + (V^2 - V_2^2).$$

Where U and U_2 are runner velocities, W and W_2 are absolute velocities of the water and V and V_2 are relative velocities of the water. This equation may be transformed into:

$2gH = AQ^2 + BQn + Cn^2$ where H = head, Q = quantity of water, n = r.p.m. and A, B and C are coefficients depending on bucket angles, friction and the proportions of the machine. This is also the general equation of the centrifugal pump with the signs changed:

$$-2gH = AQ^2 - BQn - Cn^2.$$

This is the equation of a hyperbolic paraboloid. The analysis has been very carefully worked out by Grunebaum in his pamphlet, Zur Theorie der Zentrifugalpumpen, Berlin, 1905. W. F. Uhl in *Transactions* A.S.M.E., vol. 34, page 418, has published a variation of Gelpke's types, which is convenient to use. He uses six types instead of eight.

The Holyoke Testing Flume.—The Holyoke testing flume grew out of the testing of waterwheels started by James Emerson at Lowell in the late sixties. In 1870 the Holyoke Water Co. invited Mr. Emerson to move to Holyoke and to establish a flume there. The present Holyoke flume built by Clemens Hersehell dates from 1883 and up to date more than 2,000 runners have been tested there. The flume is adapted for testing runners from 27 to 42 in. in diameter under a maximum head of about 16 to 17 ft. Capacities up to 250 cu. ft. per second may be used which reduces the available head to about 10 ft. Quantities higher than this cannot be measured with accuracy.

The accuracy and application to general practice of the Holyoke tests have been attacked very often, especially where a high-head turbine has been tested under the Holyoke conditions.

However, at the present time these tests hold "a position of generally accepted reliability" and it is the opinion of many engineers that the results shown in the flume may be bettered in a careful field test (see Larner's paper). Testing in place is now quite common and when proper precautions are taken may be quite accurately done.

The Holyoke test sheets usually contain 60 to 70 tests of 3 to 5 min. duration from which the following data is secured: number of experiment, percentage of gate opening, percentage of full discharge of wheel, head, duration of experiment, revolutions per minute, quantity of water discharged by wheel, horsepower developed, efficiency of wheel.

From these data it is possible to construct a set of characteristic curves covering the whole field of the operation of the turbine. Mead has shown this in Fig. 295 and 247 of his "Water Power Engineering." Fig. 247 is reproduced here. In this method the discharges per second under 1-ft. head are used as abscissæ with r.p.m. under 1-ft. head as ordinates. The efficiency under each gate opening and speed is plotted in its proper place with the above coördinates and the curves of equal efficiency are then drawn in. If hp. at 1-ft. head is now marked on each of the plotted points curves of equal horsepower may be drawn. From these curves the entire performance of the runner under all circumstances may be seen.

Mechanical Details.—The standard wheel consists of two crowns between which the buckets are placed. The buckets are either made of formed steel plates placed in the molds and cast into the crowns or they may be made of the same material as the rest of the runner in which case the mold is built up with cores. Cast wheels are practically universal now. No attempt at finishing is usually made except to chip or file off the fins left by the core junctions and some of the very highest efficiencies have been obtained from runners in this condition proving that surface friction does not play nearly as large a part as was formerly supposed. Careful design and good workmanship usually go together in which case much hand-finishing is not necessary. Balancing is not as necessary as in steam turbines but is done to some extent especially on tangential wheels where the buckets are usually of forged or cast steel bolted to the rim of the wheel. The larger diameter wheels are usually built on the tension-spoke principle.

Clearances between fixed and moving parts in well-built wheels are not larger than 1/16 in., although with careful design the water wasted through a clearance as large as 3/8 in. would not be excessive when the wheel is running. When the wheel is not running the leakage is determined by the kind and tightness of the gates. It is always best to have a valve in the penstock just above the wheel so that when not in use the wheel may be drained and leakage prevented. For this purpose butterfly valves of very heavy design, actuated by power have been found valuable but the best valve for low and medium heads is a plain gate valve actuated by a hydraulic cylinder. These valves should not be supposed to take the place of the stop logs or other headworks gates, but should be in addition to them.

Casings.—The casings of water turbines may be of almost any form and a good designer may display his individual taste to the full. With open penstocks the casing need only consist of the guides and the two narrow crowns holding them in place. With the closed penstock the scroll or spiral casing when well designed is the best. Cylindrical casings are used to a large extent especially when two or more wheels are on one shaft and when the casing supports the bearings in horizontal wheels. In designing casings the attempt should be made so to proportion the casing that at the loading at which the turbine is most often used the water will be brought to the guides at the required velocity without eddys and with a smooth stream flow.

Draft Tubes.—Draft tubes should be so designed that there is a uniform reduction of velocity from runner to tail water. The upper end of the draft tube should be the same diameter as the outside of the runner with a little flare at this point in the case of large-capacity runners to take care of the outward discharge from the buckets. From this point the tube should enlarge consistently to the discharge point and a flare of 1 ft. in diameter to 3 ft. in length is the maximum that may be allowed. One foot in 4 or 5 is much better. Knowing the height of the tube, the outlet is known and the loss of head can be calculated. This loss may vary from less than 1 per cent. of the total head to as much as 12 or 15 per cent. in poorly designed installations.

Gating.—Turbine gates for regulating the amount of water entering the runner are of three kinds:

1. Cylinder gates in which a thin cylinder moves axially in the clearance space between guide vanes and runner blocking more or less of the breadth of the passages. This gate leads to eddys and inefficiency unless cross-partitions are cast between the vanes of the runner to control the formation of eddys (Fig. 297).

2. Register gates in which the thin cylinder in the clearance space is perforated with holes to correspond to the guide passages and is moved circumferentially to control the size of the discharge openings in the guide passages. These gates are not used very much since they increase the friction and eddy losses seriously (Fig. 298).

3. Wicket gates in which the guide vanes are pivoted and opened or

closed to admit more or less water to the runner. This method also creates eddys at all positions but one, but they can be better controlled by this construction and wicket gates are adopted for the better class of turbines (see Fig. 299).

Regulation.—The problem of waterwheel regulation is much more complicated than the regulation of gas or steam prime movers. The

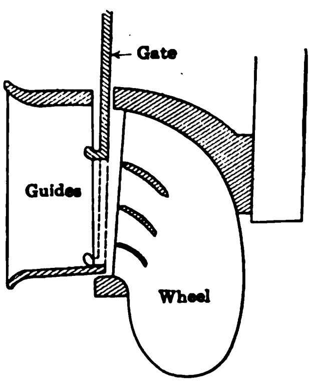


FIG. 297.—Cylinder gate.

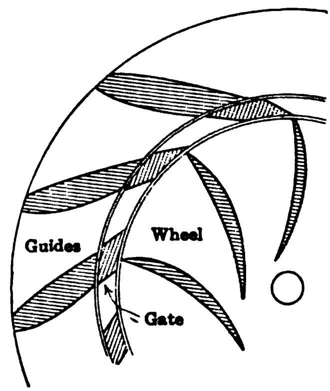


FIG. 298.—Register gate.

steam engine, with its two or more maximum impulses per revolution, presents a very simple problem, since a medium-weight flywheel can store enough energy from impulse to impulse to maintain a nearly uniform rotation until the cutoff or throttling governor can control the size of the next impulse. In a slow-moving single-cylinder engine the flywheel

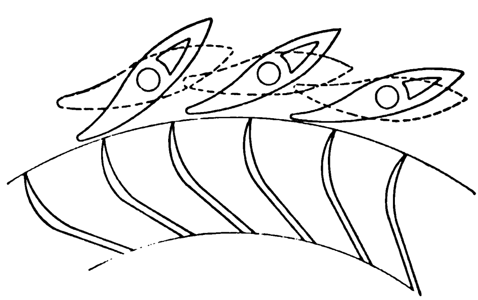


FIG. 299.—Wicket or Fink gate.

has only to store energy for $\frac{3}{10}$ sec. The regulating machinery is light in a throttling governor weighing only a few pounds, and a very small simple flyball governor furnishes sufficient power for quick and efficient working.

With the steam turbine the same small and light machinery will do work if a throttling governor is used. When nozzle governing or puff governing is used the relay principle is introduced. Here, as in the steam engine, the action is practically instantaneous and the flywheel effect of turbine and generator is always sufficient for good regulation.

A single-cylinder, single-acting, four-cycle internal-combustion engine presents a much more difficult problem. With 75 r.p.m., the impulses are a little over 2 sec. apart and the loads can vary much in 2 sec. Four cylinders would bring the interval down to ½ sec. and four double-acting cylinders to ¼ sec., but in this case the mechanisms to be moved are large and heavy, as are the friction and inertia. It is possible to design a flyball governor having sufficient power to operate them, but the oil relay is easier, cheaper and more certain.

In these types of prime movers the working fluid is very light even at high pressure and the inertia of the moving fluid is negligible, but when

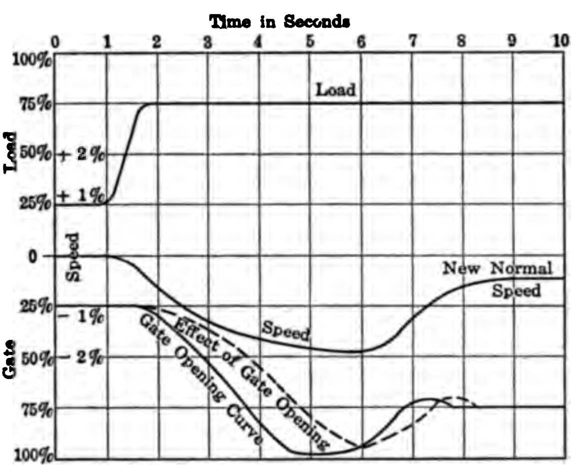


FIG. 300.—Regulation curves.

dealing with water as a source of power the weight of the fluid is the source of energy and the governing mechanisms must be strong enough to take care of the heavy shocks due to the inertia of the moving water. The gate mechanisms of turbines are usually larger than the wheel itself and generally as heavy, while the connecting links are also very heavy. All of this heavy machinery cannot be put in motion, or stopped quickly, but luckily the flywheel effect of the revolving parts is usually large and the sudden impulses are not of the order of those in steam or gas engines.

With small wheels and low heads a relay governor is used, but with higher heads or large wheels the relay does its work through a third mechanism, usually a rack and pinion. The usual arrangement includes a flyball governor which actuates two ratchets on a bar which is reciprocated continually by the relay. When the speed rises, one of the ratchets is brought into play on the ratchet wheel and the reciprocating motion moves the wheel around, tooth by tooth, closing the gate. There is always a small speed variation at which no motion of the gate follows.

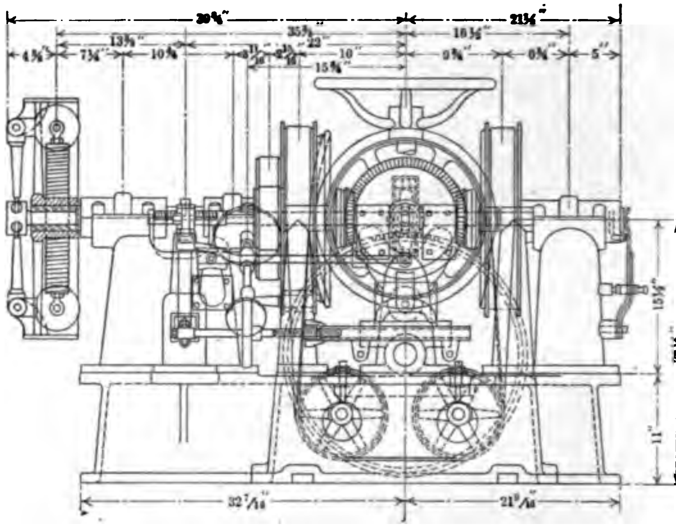


Fig. 301.—Waterwheel governor, Sanitary District of Chicago.

In this kind of a governor there is always a lag and the length of lag is adjusted to take care of some of the inertia of the water column. Much trouble in the earlier governors was caused by the attempt to run

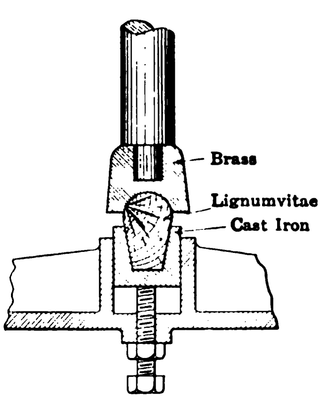


Fig. 302.—Step bearings.

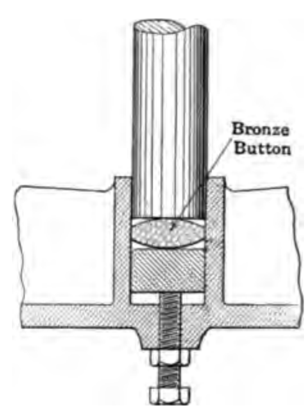


Fig. 303.—Step bearings.

them without lag and trouble also results if the lag is too much. The correct amount seems to depend on head, revolutions, size of penstock and weight of gate mechanism.

Two general principles may be stated: first, the gate must be opened only as fast as gravity can supply the water to the wheel; and second, the gates must be closed so slowly that no serious strains will be developed in the penstock from the inertia of the water. Wherever it is possible, the gate mechanism should be in static balance, except as to friction. This is not always possible, but it can be partially done in every case and the governor is then relieved of work which it should not be called upon to do. Fig. 300 shows the curves of opening and closing of a modern type of governor, and Figs. 301 and 304 show the governors

FIG. 304.—Waterwheel governor.

and the relay mechanism. For parallel running of waterwheels and steam or gas engines it is essential that the flyball governors of the machines have the same characteristics, although there are modifying influences, such as flywheel action. The difficulty of dividing the load between water and steam units has been largely overcome, and with the modern governors the troubles experienced are small.

Practically the only auxiliaries of a waterwheel, besides the governor are the thrust bearing and oiling system, and on the horizontal shaft turbines thrust bearings are usually not required. For small wheels the old lignum vitæ step bearing or phosphor-bronze button, run with water lubrication, is still the best construction, see Figs. 302, 303, but shafts larger than 5 in. and very fast-running shafts do not work well unless the design is ample. In the larger modern designs the thrust bearing is placed near the top of the shaft below the generator, where it may be supported from the floor framing, or with the later Kingsbury thrust the bearing is placed above the generator on the spider which supports the upper bearing. The older thrusts consist of two cast-iron surfaces, face to face, the lower one fixed and the upper revolving with the shaft. To the center of pressure of these surfaces oil or water is piped under sufficient

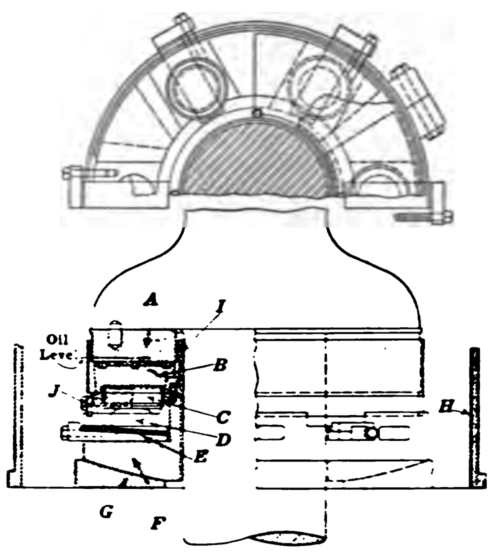


FIG. 305.—Kingsbury thrust bearing.

pressure to raise the shaft one or two-thousandths of an inch. The oil or water escapes through the orifice and the upper shoe will run on the film of fluid so made. In the Kingsbury bearing the revolving surface is replaced by a number of smaller rectangular shoes, supported at a point behind the center of pressure. These shoes are babbeted on the face side and scraped to a true surface, but the forward edges are eased to allow the oil to enter under them. The lower shoe is covered with about 3 in. of oil which may be circulated to keep it cool. These bearings are shown in Fig. 305 and Fig. 306 shows the oil-pressure bearing for the same weight, such as was used at Niagara Falls. Where the pressure bearing is used a rather complicated oil- or water-supply system is necessary and pressures up to 700 or 800 lb. per square inch are often used.

The oiling systems are usually gravity systems, provided with a sump for the dirty oil, from which it is pumped through filters to supply tanks near the roof of the station. If the head necessary to work the governor relays is higher than the gravity head in the station, two small pumps and an accumulator are usually installed for this purpose. On horizontal-shaft machines of small size, ring oiling bearings are usually used, but lignum vitæ bearings with water lubrication are also common. For the larger horizontal shaft turbines the regular gravity oiling system should be installed. Flood lubrication, with higher bearing pressures, surface speeds and oil temperatures are the tendency at the present time.

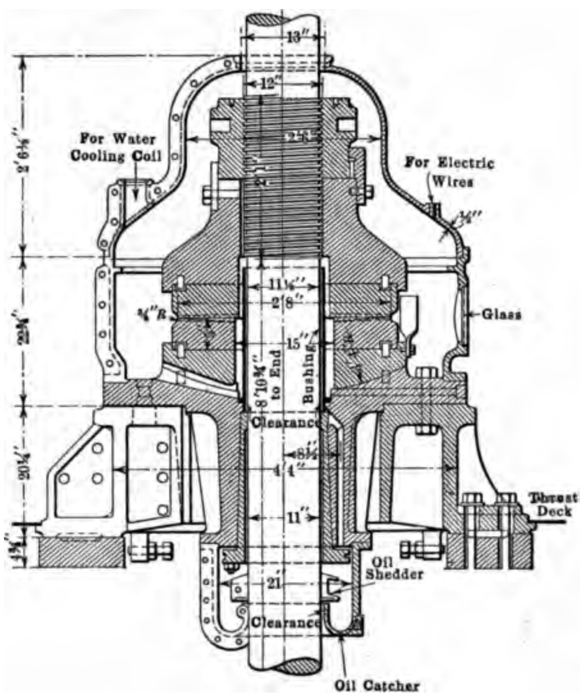


FIG. 306.—Oil-pressure thrust bearing. Niagara Falls Power Co.

Head Races, Canals and Flumes.—It is frequently necessary in order to take advantage of the total fall, to place the dam at the narrowest portion of the stream where good foundations are available and locate the power house a considerable distance down stream bringing the water to the power house in open channels, such as head races, earth or rock canals, or tunnels or pipe lines. Wooden flumes of rectangular section were formerly used for small powers or in localities where lumber was very cheap. The sides and bottom were usually made of two thicknesses of planking with broken joints. The deterioration in these flumes was very rapid.

When the canal is in earth or rock it usually pays to line it with concrete. A velocity of 2 ft. per second may be allowed in unlined earth canals and 8 ft. per second in lined canals. It is not unusual to find these canals exceeding 6,000 ft. in length. In all cases the head lost in the canal must be more than made up by the lower location of the power house.

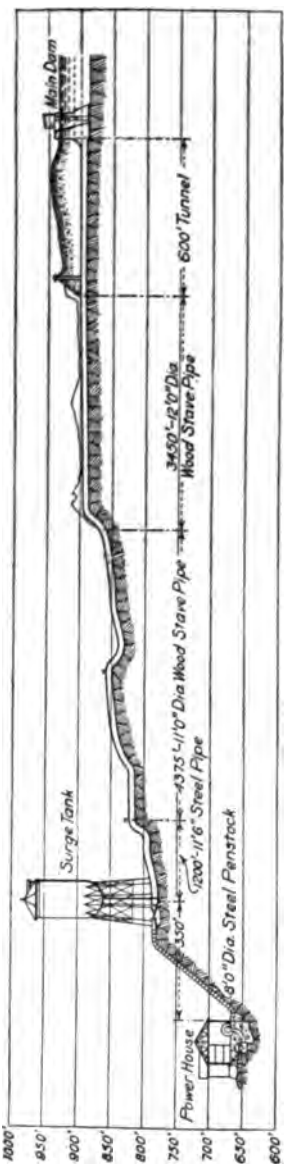


FIG. 307.—Section of Salmon River Hydro-electric Plant.

From both canals and flumes penstocks of wood or metal must be used to convey the water to the wheels or the canal or flume may be replaced by metal or wood stave pipe. The local configuration of the country usually determines which is the better plan to be used. If the stream is terraced, the canal may be the much better method.

If rock ridges exist between the dam and the proposed power house site with a winding stream and broken country, pressure tunnels in the solid rock have been driven in a straight line to the wheels, thus saving much distance and consequent loss of head. Water from a dam in one watershed has been taken in a tunnel under the ridges to the adjoining watershed and utilized to better advantage there. The lining of these tunnels should be very smooth. Very good results are obtained from this construction. The reports of the Board of Water Supply of the City of New York contain much interesting information, but the best data on these tunnels is to be found in the report on the Hetch-Hetchy Water Supply for San Francisco by John R. Freeman, Past President, A.S.M.E. This report is a model of its kind and will repay careful study by the engineer.

For figuring the slope and loss of head in canals, flumes and tunnels the tables given in Williams and Hazens "Hydraulic Tables" (Wiley) are most convenient.

Wood pipe should not be painted but, for a good length of life, should be covered in with earth (not a very good plan as then the pipe cannot be inspected) or protected by a house. Steel pipe is protected by the Augus Smith covering and either buried or unprotected. All pipes should have a large air vent near the intake end to prevent collapsing when the water is drawn off and on long pipes additional vents are often installed. Air valves should be placed on high points. The plant at Salmon Falls, N. Y., illustrated in Fig. 307, is a good example of the use of tunnel, wood pipe and steel pipe.

Water Tower or Standpipe.—If an open penstock is possible, this form of construction will undoubtedly be the best as fluctuations of head or pressure due to inertia will be very small indeed. With canal construction and low heads the surplus water can easily be taken care of by a small spillway. When closed penstocks of any length are necessary, the inertia of the moving water, as its velocity fluctuates due to the governing of the wheels to meet changes of load, may create dangerous pressures or surges. In a small installation at low head or even up to 200 ft. it has been usual to place a standpipe as close to the turbines as possible with its height great enough to reach above the dam level. In addition to this spring loaded relief valves are always furnished at the end of the penstock to further relieve the undue rise in head. The water tower may be of the plain cylindrical type as at Trenton Falls or the two diameter or differential type as at Salmon Falls (Fig. 307). They may be built of steel or reinforced concrete up to 200 ft. high, but usually the cheaper and stronger construction is steel above 100 ft. high.

A valuable chapter on the design and use of the water tower may be found in D. W. Mead's "Water Power Engineering" and a discussion of the formulas and methods of design in Parker's "Control of Water."

Speeds of Turbine and Generator.—In most cases hydro-developments are utilized by means of electric generation and transmission and it becomes important to place the r.p.m. of the turbine at such a figure that the electric generator will be reasonably cheap and suited to the service. Francis wheels may be built for many permissible speeds, in fact for any horse-power and head combination the range of speeds is very large. Table A has been calculated for $C = 90$ to $C = 10$ and for heads from 100 to 600 ft. giving minimum and maximum speed values for a number of generators from 200 kw. to 10,000 kw.

The figures under $C = 10$ are the lower speed limits for that head, under $C = 90$ the higher limits. It is well not to approach either limit too closely.

Most of the hydro-systems in this country have been designed to use 60-cycle current but a few plants have been built using 40 cycles and a considerable number using 25 cycles. In Europe other frequencies are sometimes used. Table B gives the speeds of 60-, 40- and 25-cycle generator, with common pole numbers. $\frac{2 \times 60 \times \text{cycles}}{\text{poles}} = \text{r.p.m.}$

TABLE A.—POSSIBLE REVOLUTIONS PER MINUTE

Kw.	Hp.	C 10	C 90	C 10	C 90	C 10	C 90	C 10	C 90	C 10	C 90	C 10	C 90
	Head	100-ft.		200 ft.		300 ft.		400 ft.		500 ft.		600 ft.	
10,000	13,800	27	243	63	567	106	954	152	1,368	202	1,818	253	2,277
5,000	6,900	38	342	90	810	150	1,350	215	1,935	282	2,538	358	3,222
2,000	2,760	61	549	143	1,287	237	2,133	341	3,069	450	4,050	566	5,094
1,000	1,380	85	765	202	1,818	335	3,015	480	4,320	635	5,715	798	7,182
500	690	121	1,089	286	2,574	474	4,266	680	6,120	900	8,100	1,130	10,170
300	413	157	1,413	370	3,330	614	5,526	882	7,938	1,164	10,476	1,464	13,176
200	276	182	1,638	453	4,077	750	6,750	1,078	9,702	1,425	12,825	1,790	16,110

TABLE B.—R.P.M.

Poles	2	4	6	8	10	12	14	16	18	20	22	24	26	28	30	32	36	40	72
60 cycles.....	3,600	1,800	1,200	900	720	600	514	450.0	400	360	327	300	277.0	257	240	225	200	180	100
40 cycles.....	2,400	1,200	800	600	480	400	343	300.0	367	240	218	200	185.0	172	160	150	184	120	92
25 cycles.....	1,500	750	500	375	300	250	214	187.5	167	150	136	125	115.5	107	100	94	84	75	42

Of Francis wheels from 200 kw. to 10,000 kw. installed in the last few years on heads up to 600 ft. the r.p.m. have varied from about 94 to 514, these being the limits in which the cost of the generator has balanced against the turbine cost. Six hundred r.p.m. seems to be a favorite speed for both small and large units of the tangential type but speeds as high as 900 and as low as 200 have been used.

In many cases the spacing of the units and design of the headworks settle the permissible diameters and speeds.

Cost of Hydraulic Installations. The total cost of a hydraulic installation may be divided into two parts: first, the dam, land damages, spillway, canals, flumes, pipe lines or penstocks and other details relating to the storage and transportation of the water; and second, the power house, turbines and electrical apparatus. The first group will usually amount to 50 per cent. of the total cost in small low-head plants, rising to 65, 70 and sometimes 80 per cent. in large high-head installations. Where a large amount of storage is constructed the storage itself may in large plants amount to 50 per cent. of the total cost.

The installation cost per kilowatt varies greatly, depending on local conditions, from $50 per kilowatt, where the best conditions prevail, to as high as $300 or more, where the conditions are not good. It is usually considered that a cost of $125 per kilowatt represents the limit of economical construction, with interest at 6 per cent. Where the capitalist will put up with a smaller return than 6 per cent., higher-cost plants may be possible. Not infrequently unknown physical conditions largely increase the cost of installation of a water power. When this occurs the project usually goes into a receiver's hands and enough capital is written off so that the work may proceed. In a number of cases this accounts for the very low reported costs of certain plants.

The construction of the dam is largely a matter of excavation and masonry or concrete. Earth excavation may be figured as low as 25 cts.

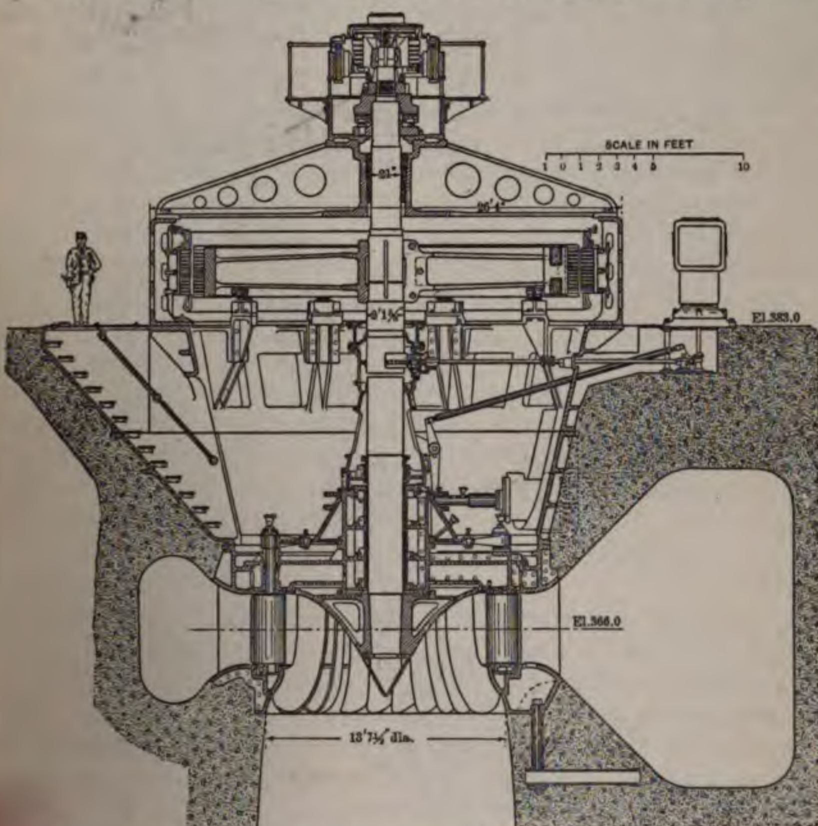


FIG. 308.—Section through 17,500 turbine unit Coosa River, Lock 12 development.

per cubic yard under good conditions, but where much hauling has to be done or difficult conditions must be met, prices as high as \$1 to \$1.25 must be used. Rock excavation is rarely cheaper than \$1 per cubic yard under the best conditions, and may go to \$3 in bad locations and as high as [illegible] in caisson work. Concrete and masonry can easily be figured [illegible] prices of sand, cement and rock are known. Concrete usually [illegible] \$4.50 to \$10 per yard in place, \$5.50 to \$7 is about the average [illegible]ncrete can be placed in mass and the forms are simple. The

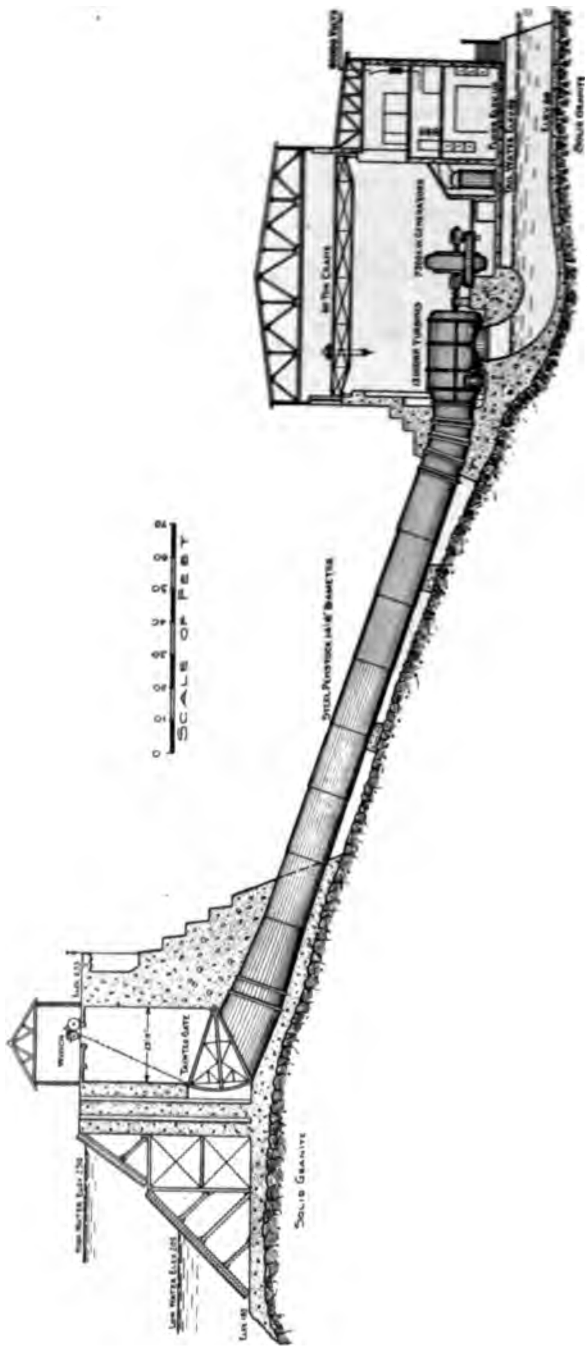


Fig. 309.—Stare Lake Power Station & Headworks, Western Canada Power Co., B. C.

local price of lumber determines the cost of the forms which will rarely be higher than $2 per yard, in which case steel forms should be considered. Lumber delivered at $40 or higher usually means that considerable steel may, with economy, be used in the forms. Cut-stone work may run from $12 to $25 per yard, but is very rarely used in commercial construction at the present day. Power houses and switch and transformer houses should be figured as in steam plants. The steel contract will average from 8 cts. per cubic foot in small buildings to 16 cts. per cubic foot in large heavy buildings, all on the basis of $80 per ton erected. The masonry contract will run from 12 cts. to 30 cts. per cubic foot depending on locality, materials and amount of terra cotta, tile and cut stone and other ornaments.

Hydraulic machinery and governors vary in price with head and size; small low-head turbines cost approximately $15 per kilowatt, while large low-head machines may be bought as low as $7. Medium-head apparatus (from 200 to 600 ft.) may vary from $13 to $7.50 per kilowatt. High-head turbines of the Pelton type run from $10 per kilowatt in small sizes down to $5 in large sizes. Generators, switchboard, exciters and cable vary from $24 per kilowatt for small low-head (low-speed apparatus) to $8 per kilowatt for large high-head high-speed machinery. Transformers cost $6 to $8 per kilowatt.

Penstocks or pressure pipes of riveted steel vary in cost from 3 cts. to 6 cts. per pound erected, plus freight and haulage. Wood stave pipe, used to such a large extent for low-head pressure pipes, will cost about 15 cts. per foot board measure, erected, with bands, for medium sizes and pressures.

Gates of the Taintor type will not usually run above 5 cts. a pound erected, plus freight and haulage. Stoney gates in small sizes may run to 8 cts., but in the large sizes should not exceed 5 cts. Cranes for use inside the power station of both the alternating-current and direct-current types, may be figured at $4.50 per ton lifted per foot of span for small short-span cranes, down to $2.50 per ton-foot for heavy long-span cranes. Special cranes, used outside the power house for handling gates, may cost anywhere from $3 to $10 per ton-foot, depending on the design.

The question of land damages, due to flooding caused by the creation of the pond, is an extremely important one. This cost in certain plants in the West, where the flooded lands were far from a settled district, have been as low as $1 per acre, while in certain of the large city water-supply reservoirs, the land damages amounted to over $250 per acre. On an average for developments reasonably removed from towns, prices from $70 to $115 per acre have been paid.

Railroads and highways usually follow rather closely the flow line of a river, and these constructions must be relocated previous to the construction of the dam. $3,000 per mile is a fair price for country highway relocation and from $5,000 to $8,000 per mile will cover the relocating of a good State road in localities where good materials are common. The cost of railroad relocation is another matter. Single-track little-used roads may usually be relocated at a cost not exceeding $60,000 a mile, but the cost of relocating a double-track express road has usually been found so high that it has not been attempted.

TABLE OF ESTIMATED COST PER HORSEPOWER OF WATER-POWER PLANTS

Having horizontal turbines, steel penstocks, and walled tailraces (dam and buildings not included)

Hp.	"L"	10 ft. fall	15 ft. fall	20 ft. fall	30 ft. fall	40 ft. fall
1,000	100	$65.14	$40.92	$29.37	$19.40	$14.60
	600	98.75	63.75	46.70	30.95	23.55
900	100	65.35	41.00	29.55	19.55	14.80
	600	98.95	63.90	46.95	31.00	23.85
800	100	65.50	41.10	29.65	19.70	15.00
	600	99.15	64.00	47.00	31.15	23.95
700	100	65.70	41.20	29.85	19.90	15.10
	600	99.50	63.95	47.25	31.35	24.15
600	100	65.85	41.55	30.00	20.00	15.35
	600	100.10	64.40	47.40	31.80	24.55
500	100	66.00	41.70	30.25	20.25	15.50
	600	100.10	64.00	47.85	31.80	24.45
400	100	66.30	42.05	30.55	20.80	16.00
	600	100.00	65.15	48.05	32.35	25.10
300	100	66.85	42.65	31.10	21.50	16.50
	600	101.00	65.80	48.50	33.20	25.65
200	100	68.50	44.20	32.45	22.60	17.60
	600	102.85	67.35	50.60	34.45	26.95
100	100	71.40	46.65	34.75	24.75	19.80
	600	106.60	70.30	52.90	36.85	30.80

"*L*" = distance from feeder head to end of tailrace, cost of canal, if any, not included.

Cost of Hydro-electric Developments.—The cost of hydro-electric developments depends upon many conditions, such as water rights, real estate, right-of-way, the cost of the development, and the distribution system. Further, the depreciation, repairs, taxes, insurance, interest on the investment, operating expenses, etc., enter into the account.

The cost of the enterprise depends very much on the character and the conditions under which the development is carried out, and the cost per unit capacity depends upon the total capacity of the plant. It occurs quite frequently that the unit cost in large propositions is greater than in small ones, although it would appear that it should be smaller.

TABLE I.—ESTIMATE OF COST OF VARIOUS DEVELOPMENTS

Location of development	Natural head	Available head	Power developed, hp.	Estimated capital cost	Cost per hp.
Healey's Falls, Lower Trent River...	...	60	8,000	$675,000	$84.38
Middle Falls, Lower Trent River...	...	30	5,200	475,000	91.37
Rauney's Fall....................	...	35	6,000	425,000	69.67
Rapids above Glen Miller..........	...	18	3,200	350,000	109.38
Rapids above Trenton.............	...	18	3,200	370,000	115.63
Maitland River[1]..................	...	80	1,600	325,000	203.12
Sangeen River....................	...	40	1,333	250,000	187.53
Beaver River (Eugenia Falls).......	...	420	2,267	291,000	128.28
Severn River (Big Chute)[2].........	...	52	4,000	350,000	87.50
South River.....................	...	85	750	150,000	153.33
St. Lawrence River, Iroquois, Ont...	...	12	1,200	179,000	149.16
Mississippi River, High Falls, "A"[3].	...	78	2,400	195,000	81.25
Mississippi River, High Falls, "B"[4].	...	78	1,100	123,000	181.82
Montreal River, Fountain Falls, Ont.	...	27	2,400	214,000	89.16
Dog Lake, Kaministiquia River[3]....	347	310	13,676	832,000	61.00
	347	310	6,840	619,700	91.00
Cameron Rapids..................	39	...	16,350	815,000	50.00
	39	...	8,250	600,000	73.00
Slate Falls......................	31	40	3,686	357,600	97.00
	31	40	1,843	260,000	141.00

[1] Dam rather expensive. [2] Headworks and canal less expensive than ordinary. [3] With storage development. [4] Including 3,500 ft. of head water tunnel.

TABLE II.—ESTIMATE OF COST OF HYDRO-ELECTRIC PLANTS AT NIAGARA FALLS

	24-hr. power capacity		
	50,000-hp. development	75,000-hp. development	100,000-hp. development
Tunnel tailraces..................	$1,250,000	$1,250,000	$1,250,000
Headworks and canal..............	450,000	450,000	450,000
Wheel pit........................	500,000	700,000	700,000
Power house......................	300,000	450,000	600,000
Hydraulic equipment..............	1,080,000	144,000	1,980,000
Electrical equipment..............	760,000	910,000	1,400,000
Transformer station and equipment.	350,000	525,000	700,000
Office building and machine shop...	100,000	100,000	100,000
Miscellaneous....................	75,000	75,000	75,000
	$4,865,000	$5,900,000	$7,255,000
Engineering and misc. 10 per cent. of above making total construction cost...........................	$5,350,000	$6,490,000	$7,980,000
Interest, 2 years at 4 per cent......	436,560	529,548	651,168
Total capital cost.................	$5,786,560	$7,019,584	$8,631,168
Per horsepower...................	$114	$94	$86

This is due to the fact that in many large propositions a heavy expense is involved in the harnessing of great volumes of water.

Table I, accompanying, gives figures on the estimated costs of various developments tabulated by the Ontario Hydro-electric Power Commission, and Table II gives the estimated cost of a hydro-electric plant at Niagara Falls, as given in the report of above-named Commission.

It will be noticed by reference to Table I that the cost of hydro-electric plants per horsepower, varies greatly (from \$61 to \$203) and may vary even more. Correct estimates can be arrived at only by thorough investigation of all the factors, considering with especial care the important element of depreciation.

In estimating the cost of power (that is, the generation and distribution, which of course depends very much upon the load factor) administration and operating expense, maintenance, depreciation, interest, insurance, etc., must also be well considered. The following table clearly illustrates the cost of power at the development of the Chicago Sanitary District System.

TABLE III.—COST OF POWER, CHICAGO SANITARY DISTRICT SYSTEM

Total cost of development and transmission........		\$3,500,000
Estimates of cost		
Interest on investment at 4 per cent...............	\$140,000.00	
Taxes on real estate, buildings, etc................	7,200.00	
Depreciation on buildings at 1 per cent............	3,650.00	
Depreciation on waterwheels at 2 per cent.........	2,027.32	
Depreciation on generators at 2 per cent...........	1,824.60	
Depreciation on pole lines at 3 per cent............	2,020.50	
Depreciation on other electrical appliances at 3 per cent....................................	3,995.52	
Total fixed charge..........................		161,137.94
Operating expenses		
Power and substation labor.......................	\$63,240.00	
Repairs to machinery and building................	3,700.00	
Incidental expenses..............................	1,200.00	
Operating Lawrence Avenue pumping station.......	43,960.00	
Operating 39th Avenue pumping station...........	120,380.00	
Interest on investment 39th Street pumping station..	15,599.76	
Total operating expense.....................		248,079.76
Total cost to sanitary district.................		\$409,217.76
Capacity, 15,000 hp., cost per hp. per annum....		26.40

A most important item in determining the cost of power is the cost of distribution. This is particularly true in long-distance transmission systems where the skill of the engineer is of vital importance in selecting the proper route and the kind of systems to be employed.

Whether a long-distance power-transmission project will pay will depend upon the cost of generating the power, the cost of transmission, the transmission loss, etc., and the value of energy at the point of distribution, *i.e.*, the cost at which energy might be generated at this point by some other system, as, for instance, by a steam power plant. The difference between the cost of power at the generating end and its value at the point of distribution represents the maximum cost of transmission allowable.

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